This Month's Cover: High speeds in machine design symbolized by Penton artists. Created by Robin Allen and rendered by A. L. MacBain, the cover dramatizes the familiar gyroscope rotor.

MACHINE DESIGN

Vol. 22—No. 4 April, 1950

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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DESIGNING FOR PRODUCTION AND SALES • STYLING
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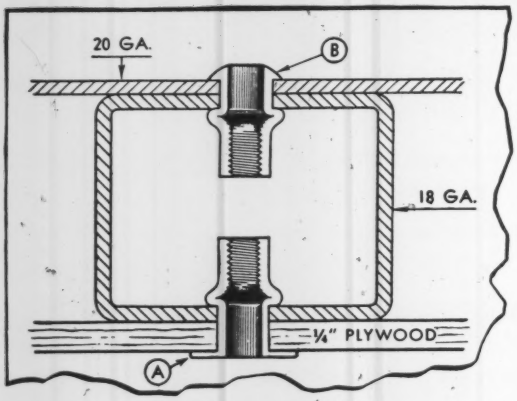
MD

NEW, BETTER WAY
to fasten plywood to steel



Trailer designers had the problem of securing ¼" plywood walls to 18 gage tubular steel sections with rivets that would not be pulled through the plywood under stress or vibrate loose in service. They found the answer in the new B. F. Goodrich Rivnut with the large washer head.

Washer-head Rivnut "A" is easier to install than either wood screws or self-tapping screws. The plywood can be countersunk so that the head of the Rivnut is flush with the plywood. Rivnuts eliminate the need for wood filler in the tubing. Brazier-head Rivnut "B" fastens either 1 or 2 sheets of 20 gage steel to the opposite side of the section.

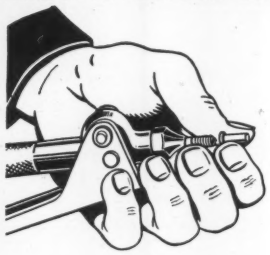
Both Rivnuts serve as blind rivets. They can also be used as nut plates for fastening trim strips and other attachments. Because installation is done by one man, from one side of the work, in a few seconds, costs for the whole operation are reduced. If you have a fastening problem, chances are there's a Rivnut that will solve it. Why not get complete information? Write The B. F. Goodrich Company, Dept. MD-40, Akron, Ohio.

THIS IS A B. F. GOODRICH RIVNUT...THIS IS HOW IT WORKS

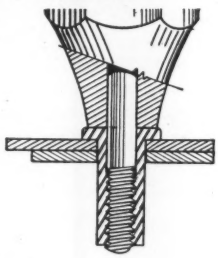
This cross-section shows the threads by which the Rivnut is threaded onto the pull-up stud of the heading tool. These threads are not injured by the heading operation and enable you to fasten *to* a Rivnut as well as *with* it.

Note counterbored portion of Rivnut in which lateral bulge is formed by pull-up action.

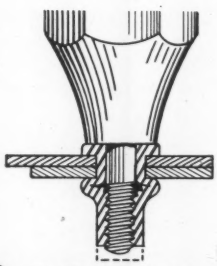
The Rivnut provides a large bearing surface, requires only a small hole, and can be installed after enameling without marring the finish.



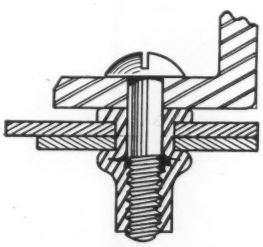
1. Rivnut is threaded onto pull-up stud of a manual or pneumatic heading tool.



2. Rivnut is inserted—head firmly against work, tool at right angles to work.



3. Tool lever operates pull-up stud, forming a bulge in the Rivnut shank.



4. After upset, Rivnut threads are still clean and intact, ready for screw attachment.

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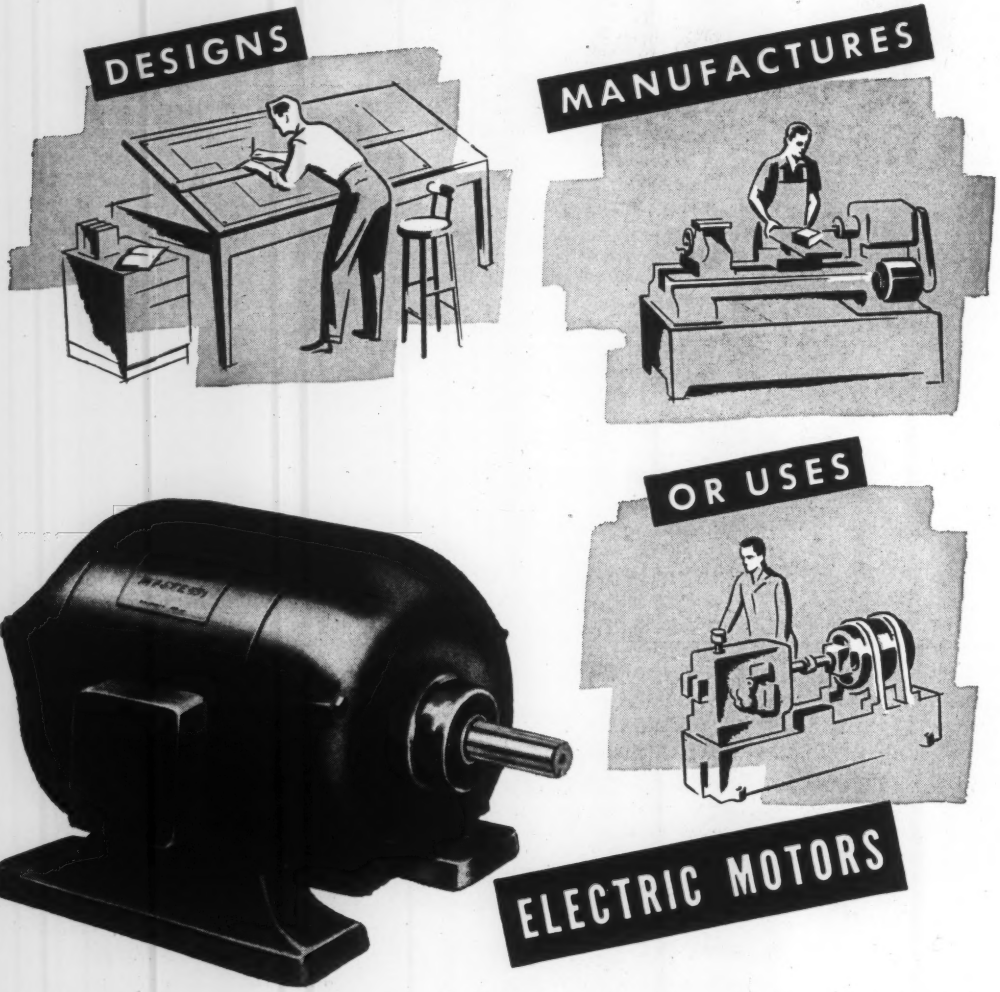

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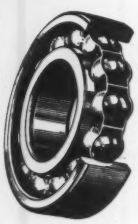
91



As almost everyone knows, many modern electric motors never need re-lubrication.

They run—an ever-increasing number of them—on New Departure self-enclosed ball bearings. For these are the bearings which (1) keep lubricant in, (2) keep dirt out, (3) reduce maintenance, and (4) simplify design.

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Yes, you can save money by being certain the electric motors you design or manufacture or use are equipped with New Departure self-sealed ball bearings. Interesting booklet "Sealed" will be mailed on request.

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MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

U. S. Slicing Machine Company has long been known for the fine engineering and precision craftsmanship employed in the manufacture of their products. Smooth operation of the gauge plate in their Model 805 Slicer is assured by a G.S. worm and shaft as well as the worm wheel assembly pictured here. Isn't it significant that wherever *really fine precision* is an essential requirement in Fractional Horsepower Gearing, that's where you'll usually find Gears by G.S? More and more of our country's largest and most discriminating manufacturers depend upon us for the production runs of the Small Gears they need. If you, too, want *better,* more *uniform, longer-lived* Gearing correctly designed and cut to exacting specifications, put *your* problems up to our competent engineers. Suggestions, ideas, and moderate cost estimates won't cost you one penny! Will you write us today?

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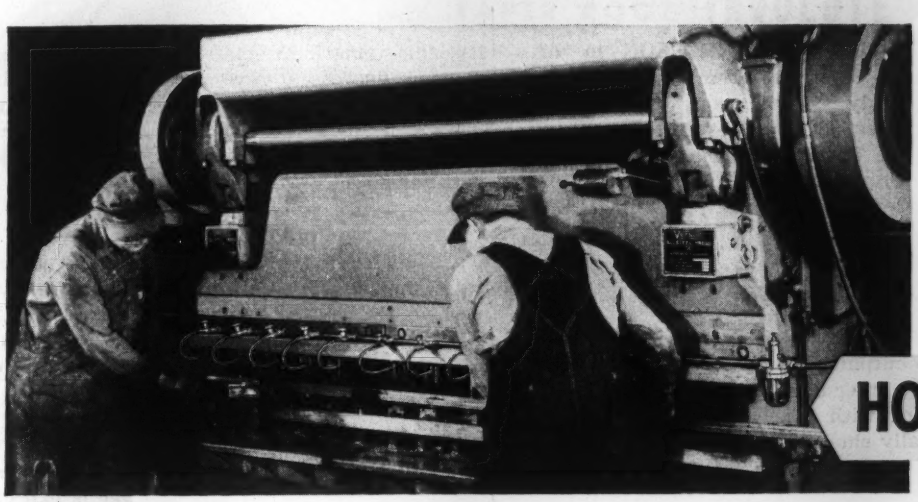

See our Exhibit
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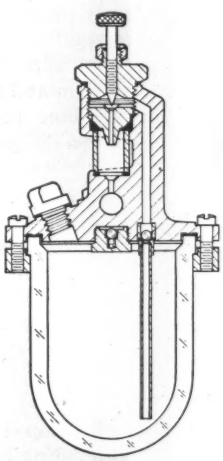
A. S. T. E. Show

Sunlight Metal Products Corp.—Denver manufacturers of quality steel sash—did the trick! Upon the advice of a Norgren representative, they installed a Norgren Oil Fog Lubricator and a simple distribution system on a punch press—to lubricate ***automatically*** each punch and stripper unit for ***every stroke.*** Previously, the punches were lubricated by hand with a brush, only once in every ten strokes.

Here's a tip for machine designers and users to give their equipment more output and less upkeep by the simple and logical installation of Norgren Automatic Oil Fog Lubricators.



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- Compact, easy to install.
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- Accurate needle-valve control of oil flow.
- Ball check holds oil in position for intermittant use, assuring oil feed the instant air flows.
- Develop an airborne Oil Fog that coats every moving part reached by air.
- Stop oil waste; lubricate only when air flows.

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TOPICS

METALLURGICAL INFORMATION in all printed or recorded forms can be found easily and quickly as the result of a recent classification index developed by ASM in co-operation with the Special Libraries Association. When used with a punch card system the classification provides for as many as 181,000 different index headings on metallurgical information.

NEW BETATRON at University of Illinois has a steady output of twenty times more electrons than any other machine has ever produced. Research work with this 300,000,000-volt machine eventually should transfer additional laws of physics from theory to practical engineering, opening new fields of design and development.

THERMOCOUPLE TABLES are being revised by the National Bureau of Standards, taking into account recent changes in electrical units and in the international temperature scale. During the year eight tables will be published for temperature-emf relations for platinum-platinum rhodium, chromel-alumel, and copper-constantan thermocouples.

PLASTIC MOLDING powders, made from cod fish protein, are now being produced in Norway. Known as Monoplast and developed by the firm Wm. A. Mohn & Son, the powders produce molded parts with exceptionally fine finish and with properties equal to parts molded from other types of powders.

BOTH MECHANICAL AND ELECTRICAL engineering degrees will be awarded for a new five-year course at Lehigh University. Recognizing that much design work involves both branches of engineering, the school has set up the course to give the mechanical degree at the end of four years and the electrical degree at the end of the fifth.

GEIGER COUNTER has been adapted by the research laboratory of the United States Steel Corp. for use as a quick and accurate method of analyzing steel samples. In addition to being capable of detecting and measuring radioactivity, this remarkable instrument is also able to measure fluorescent rays, each of a special kind, that emanate from steel when its atoms are ionized by powerful X-ray. Analysis with a Geiger counter is, in many ways, simpler than with the direct reading spectrograph.

STABLE GRAPHITE FILM may be successfully applied to practically any surface by a new process developed by Electrofilm Corp. Applied by spray or dip, the 0.00015 to 0.0005-inch film has excellent resistance to abrasion and exceptional bearing strength.

EXPERIMENTAL ATOMIC POWER plant for generating useful electric power is expected to be in operation at the Knolls Atomic Power Laboratory within the next two or three years, according to the General Eletric Co. Several decades, however, may be required before significant quantities of power are generated from nuclear fuel.

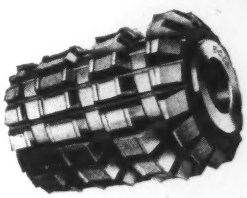
TITANIUM ALLOY, lightweight and high strength, has been developed by the Navy Bureau of Aeronautics. It retains basic properties at high temperatures. Highly resistant to corrosion, the new alloy, composed basically of 5 per cent chromium, 3 per cent aluminum and the rest titanium, is being used in jet-plane parts where strength at high temperatures is most important.

CAST CERAMIC material, five to ten times stronger than other cast ceramics, has been developed by the Air Force. It has a very low dielectric constant and tests are under way to determine its exact properties.

ELECTRONIC WATCH TIMER, developed by Salford Electrical Instruments Ltd., England, compares the ticks of a watch with a standard frequency, providing a rapid check of the watch's accuracy. The standard timing circuit, supplied by a crystal-controlled oscillator and a series of frequency dividers, causes a spot to rotate on the screen of a cathode-ray tube. Watch is adjusted correctly when its spot on the screen appears to stand still.

Solve THESE DESIGN PROBLEMS

ON POWER TRANSMISSION DRIVES *with* TAPER ROOT INVOLUTE SPLINES



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Taper Root Involute Splines seldom require a length more than equal to the pitch diameter, which is sufficient for carrying any potential load.

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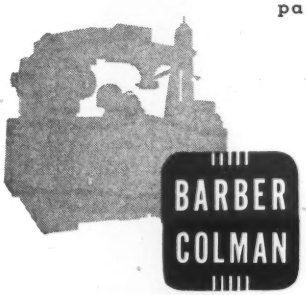
Taper Root Involute Splines offer about 50% more root contact area than parallel key mounts. Tests prove the strength of this Design equals that of a solid shaft, slightly larger in diameter than the root diameter of the spline.

3 EASY MACHINING, LOW TOOLING COST
Taper Root Involute Spline Hobs have more teeth, remove less metal per tooth space, and run at faster RPM's than parallel key hobs. Tool life is longer because clearance lugs are reduced.

4 LOCATING SHOULDERS ELIMINATED
Taper Root Involute Splines locate against the taper and are thereby self-centering. Equalized bearing and stresses result on all teeth.



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GENERAL OFFICES AND PLANT, 7714 LOOMIS STREET, ROCKFORD, ILLINOIS, U.S.A.

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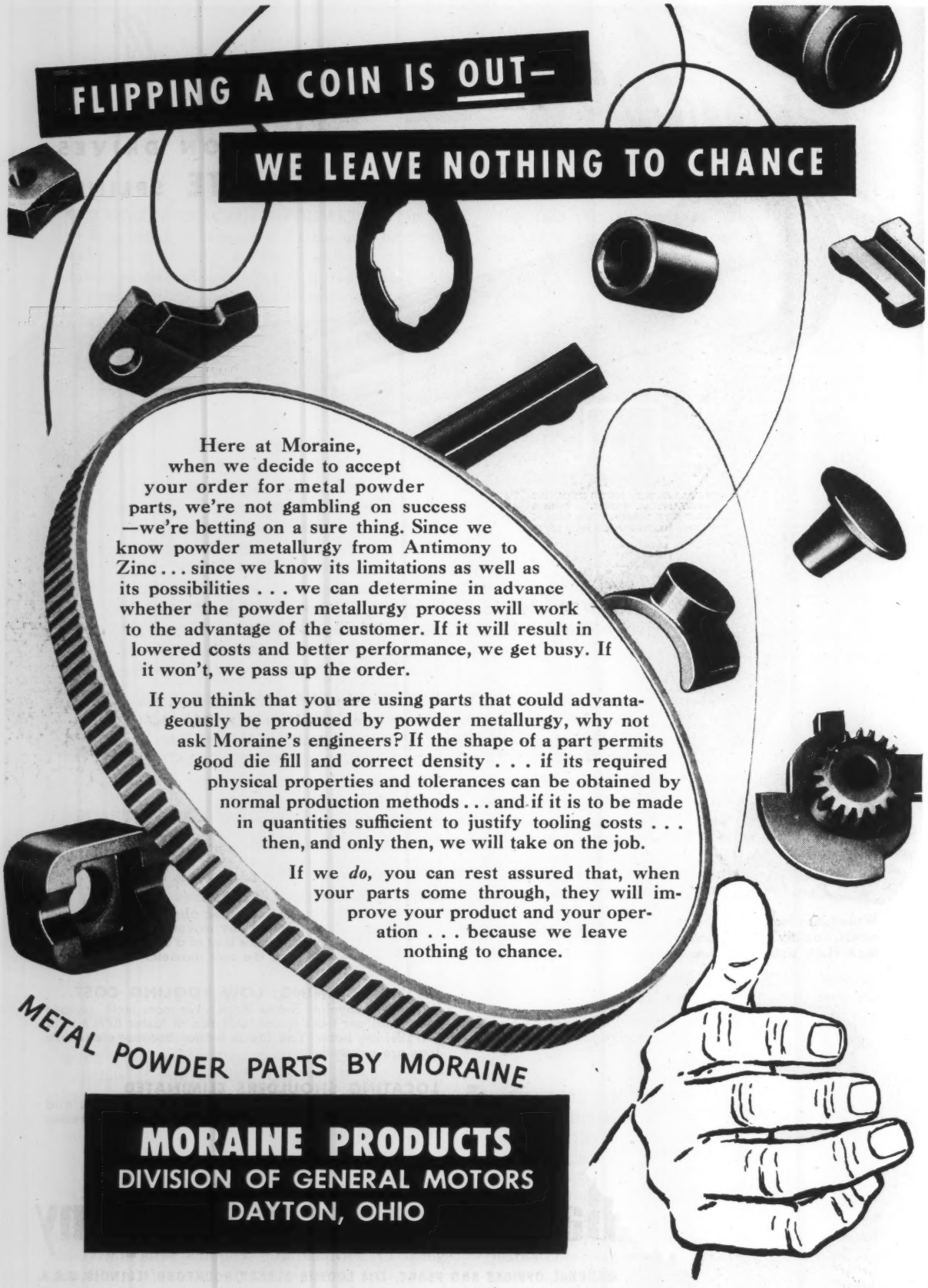


Here at Moraine, when we decide to accept your order for metal powder parts, we're not gambling on success —we're betting on a sure thing. Since we know powder metallurgy from Antimony to Zinc . . . since we know its limitations as well as its possibilities . . . we can determine in advance whether the powder metallurgy process will work to the advantage of the customer. If it will result in lowered costs and better performance, we get busy. If it won't, we pass up the order.

If you think that you are using parts that could advantageously be produced by powder metallurgy, why not ask Moraine's engineers? If the shape of a part permits good die fill and correct density . . . if its required physical properties and tolerances can be obtained by normal production methods . . . and if it is to be made in quantities sufficient to justify tooling costs . . . then, and only then, we will take on the job.

If we *do*, you can rest assured that, when your parts come through, they will improve your product and your operation . . . because we leave nothing to chance.

METAL POWDER PARTS BY MORAINE

MORAINE PRODUCTS
DIVISION OF GENERAL MOTORS
DAYTON, OHIO

MACHINE DESIGN
A PENTON PUBLICATION

The Challenge of High Speed

WITHOUT doubt some of the most intriguing problems in machine design are those concerned with matching a source of power to a task which a machine is to perform. These problems are so fundamental that MACHINE DESIGN has, for a dozen years, devoted each April issue to machine drives and controls.

In planning these issues the editors have striven to present a well balanced symposium of contributed articles on mechanical, electrical, electronic, hydraulic, and pneumatic drives and controls. The objectives were twofold—to report significant developments and to review the current state of the art.

This year is no exception, but something new has been added. Some months ago the editors took a fresh look at the needs of their readers. As they delved deeper into the whole field of machine drives they became aware of the great dearth of design information on high-speed machine drives. Successive checks of the various types of drives—gears, belts and chains, as well as hydraulic, pneumatic and electric motors—revealed a uniform lack of real engineering data for any but conventional speeds.

And so was conceived the special 48-page section which appears in this issue. Roger Bolz and his able collaborators Jack Greve, Bob Harnar and Bob Denega, have scoured the highways and byways for practical information that an engineer can apply in actual design. Their efforts have resulted in a comprehensive guide to the design and selection of drives — both primary and intermediate—for obtaining speeds beyond conventional. It consists of a series of integrated articles each dealing with a particular type of drive, preceded by a quick review of the fields of application of each. No attempt is made to tell the designer what drive to choose, but the information is set down in such a form that the reader will have no difficulty in evaluating each in terms of his own particular problems.

Although certain specialized requirements prompted the development of many of the drives discussed, their availability now opens up new and exciting possibilities to the designer. While certain physical characteristics appear to prescribe top limits to the speed of each type of drive discussed, these are not to be taken as absolute. For instance, as we go to press we learn that recent experimental work with high-speed gears has pushed the possible limits beyond those indicated in our presentation. The limits shown for each drive should therefore be regarded as a challenge rather than an obstacle and should inspire designers to extend these limits even farther.

Colin Carmichael

EDITOR

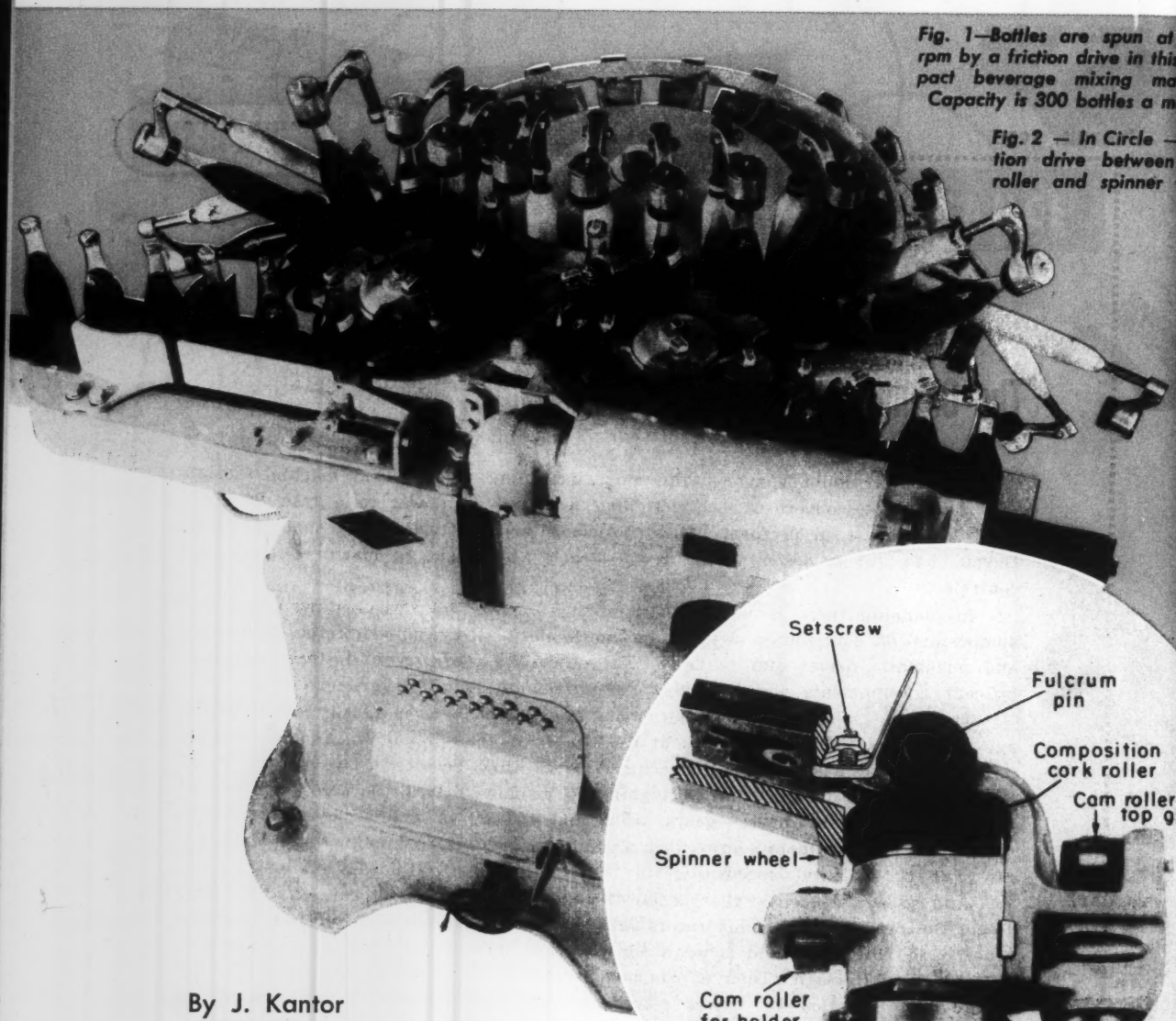
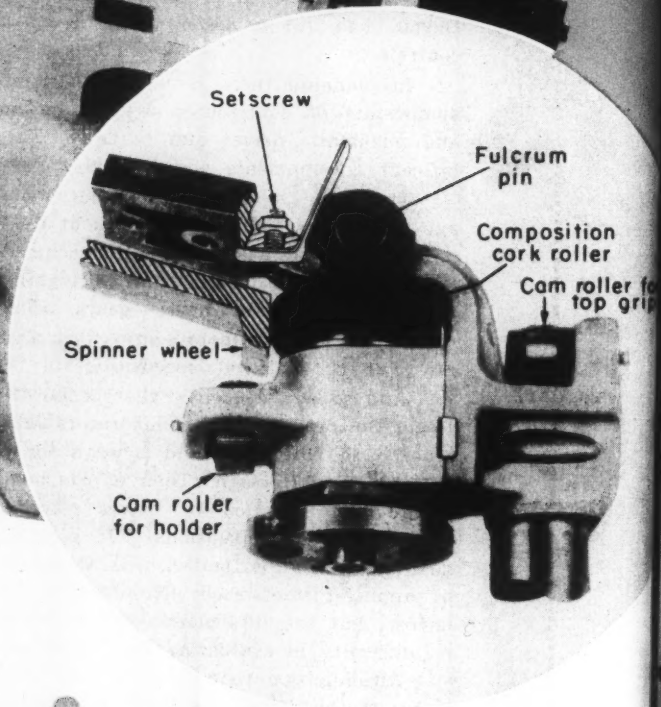


Fig. 1—Bottles are spun at 20[illegible] rpm by a friction drive in this co[illegible] pact beverage mixing machi[illegible] Capacity is 300 bottles a minu[illegible]

Fig. 2 — In Circle — Fri[illegible] tion drive between [illegible] roller and spinner wh[illegible]



By J. Kantor
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Friction Drive

. . . solves design problems in beverage mixer

IN THE design and development of bottling machinery, high production at low cost is now, as always, the ultimate objective. Thirty-seven years ago, one bottling machine was capable of handling only twenty-four bottles per minute, whereas a modern machine will process three hundred bottles per minute. All auxiliary machines should be capable of handling bottles at the speed of the filling machine.

In carbonated beverage bottling machines, the syrup and carbonated water are placed unmixed in the bottle to prevent excessive foaming. The bottles are capped and then processed to thoroughly blend the water

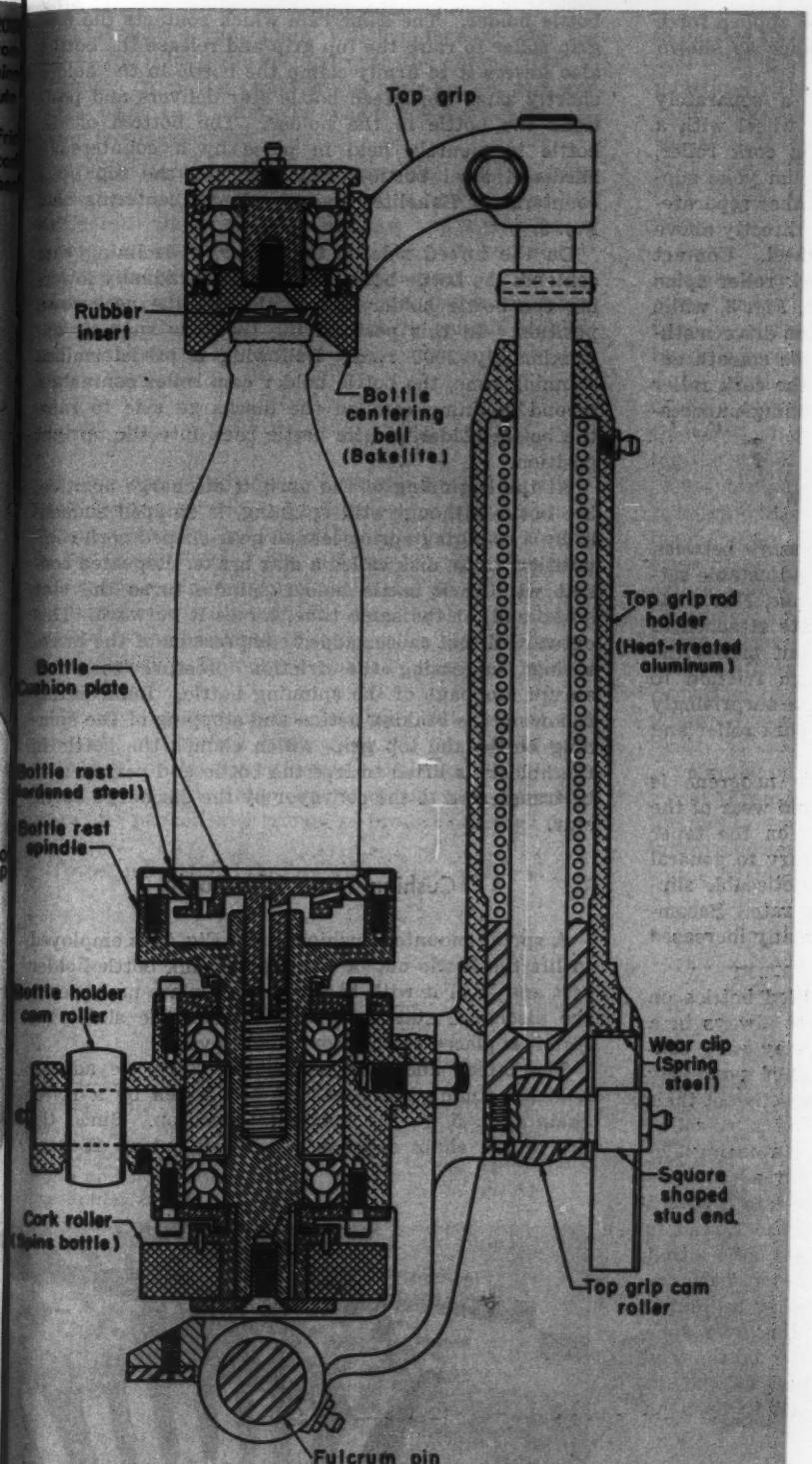


Fig. 3—Bottle holder showing method of clamping bottles, cam rollers for inverting and releasing bottles, and cork roller for spinning them

and syrup. The machine used for mixing the bottled beverage (syrup and water) is commonly known as a mixer. Various types of mixers incorporating different principles of mixing have been developed. The most recent design is called a spinner mixer, *Fig.* 1, to which this article will be confined. It is interesting from the engineering standpoint because of the simple solution to many design problems which at first appeared to involve mechanical complications.

Is Based on Simplicity

In designing any commercial machine intended for quantity production and competitive sale, determination of its size becomes an item of importance. While the required capacity of the machine determines this to a large extent, the intended selling price is also a controlling factor. As the size increases, so does the manufacturing cost and eventually the selling price. Good design based on simplicity of parts and low-cost production, therefore, plays an important part in the size and its relation to selling price.

BOTTLE SPINNING MEANS: In the development of the spinner mixer, the principle of spinning the unmixed, bottled beverage at speeds approaching 2000 rpm in the neck-down or inverted position had been established as being the most effective. The most perplexing problem in the basic conception of the mixer was to decide on the best and simplest method of imparting a spinning action to the filled bottle in the neck-down position. Again, by what simple means should the bottle holder be engaged and disengaged from the drive imparting spinning action to the bottle holder? In the final analysis, it became apparent that a friction drive would be the simplest and best suited for smooth engaging and disengaging.

In any friction drive, it is desirable to design so as to make possible adjustment of pressure between the driven roller and the drive wheel. This problem suggested the principle of levers, making use of a third-class lever in which the applied force (weight of bottle holder) and the resistance

are on the same side of the fulcrum, the applied force being further from it than the resistance as shown in the drawing in *Fig.* 3.

The driving means developed into a separately driven cast iron disk called a spinner wheel with a tapered periphery to contact a friction cork roller, *Fig.* 2, of the bottle holder. The fulcrum yoke supporting the bottle holder bolts onto another separately driven revolving disk called a spider directly above the friction-drive cast-iron spinner wheel. Contact between the spinner wheel and the cork roller spins the bottle clamped in the bottle holder, *Fig.* 3, while in the neck-down position. This friction-drive method of spinning the bottle makes possible smooth engagement and disengagement between the cork roller and cast iron spinner wheel without adding unnecessarily complicated mechanisms.

Setscrew Adjusts Pressure

To enable control of the bearing pressure between the cork roller and spinning wheel, an adjustable setscrew is provided in the fulcrum pin yoke, *Fig.* 2. It bears against a hardened-steel wear plate attached to the aluminum bottle holder. A jam nut makes the adjustment secure. To provide enough friction to spin the bottle at a rate of 2000 rpm, a surprisingly small pressure is required between the cork roller and the spinner wheel.

Granulated cork bound together by neoprene is utilized for the roller. To overcome rapid wear of the cork, a thin film of oil is maintained on the taper periphery of the spinner wheel. Contrary to general belief, the oil film does not cause any noticeable slippage or reduction of the bottle spinning rate. Becoming saturated with oil, the roller has greatly increased life.

BOTTLE TRANSFERRING: Since the filled bottles on the conveyor feeding the mixer are not always in a solid line, the spacing between them may vary considerably. This presented a problem of positively feeding bottles into the infeed star irrespective of their spacing. It was successfully solved by a so-called bottle infeed worm, *Fig.* 4, resembling a rotating cylinder having a coarse spiral tapering thread contacting the bottle and guiding it accurately into a pocket of the infeed star. The beginning of the thread is shallow and narrow to fit in between two bottles that are touching each other. The thread toward the opposite end is wider and deeper. A shallow beginning also prevents a single bottle from toppling over upon contact with the thread. Opposite the bottle feed screw a bottle guide is flexibly mounted so that, in the event of a bottle jam, it moves out to open an electric circuit and stop the mixer.

Guides Prevent Jamming

At the entrance to the bottle feed screw the bottle guides are curved to prevent a bottle that has toppled over on the conveyor from being fed into the feed worm, resulting in a jam.

An infeed bottle star transfers the filled bottle from the conveyor, *Fig.* 4, and positions it accurately in the bottle holder. The same cam which contacts the top-grip roller to raise the top grip and release the bottle, also lowers it to firmly clamp the bottle in the holder shortly after the infeed bottle star delivers and positions the bottle in the holder. The bottom of the bottle is securely held in place by a countersunk hardened steel bottle-holder rest and the top by a countersunk, Bakelite retainer called a centering bell, *Fig.* 3.

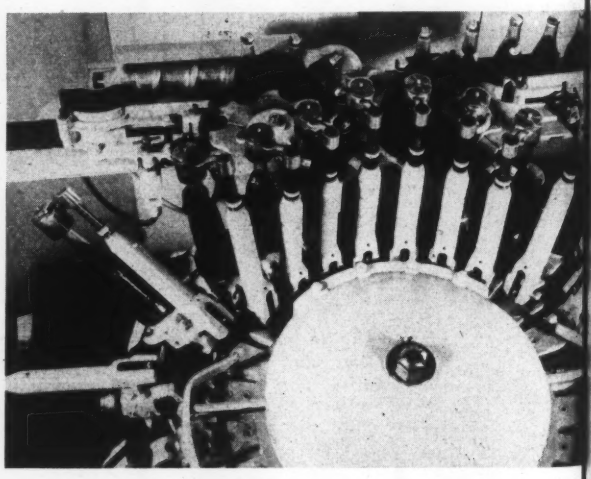
On the infeed side of the mixer, a declining cam contacts the bottle-holder cam roller, gradually lowering the bottle holder and bottle into the neck-down position. In this position the bottle is spun at approximately 2000 rpm. Following a predetermined spinning time, the bottle holder cam roller contacts a second inclining cam on the discharge side to raise the bottle holder and its bottle back into the upright position.

At the beginning of the upright discharge position, the bottle although still spinning, is stopped suddenly by a swinging spring-loaded gear-shaped cork-composition brake disk called a star brake. Repeated contact with each bottle holder spindle turns the star brake and, at the same time, forces it outward. This outward thrust causes added compression of the brake spring, increasing the friction pressure to effect abrupt stoppage of the spinning bottle. Immediately following the braking action and stopping of the spinning bottle, the top grip, which clamps the bottle in the holder, is lifted to free the bottle and permit it to be transferred to the conveyor by the discharge bottle star.

Cushion Plate Lifts Bottle

A spring-mounted cushion plate, *Fig.* 3, is employed to lift the bottle out of the countersunk bottle-holder rest and level it with the stationary plate underneath the discharge star, over which the bottle slides before it is placed on the moving conveyor.

The infeed and discharge bottle stars are adjustable for timing purposes and are driven by a roller chain, *Fig.* 5, in the horizontal position. Since the life of the chain depends to a great extent on good



lubrication, a chain tightening idler sprocket is built integral with a vane pump which pumps oil over the top of the idler sprocket and onto the chain. The oil dripping from the chain drops into an oil pan and flows back to a sump in which the lubricating oil pump is submerged.

BOTTLE HOLDER: At high operating speeds of 300 bottles per minute of 7-oz capacity, lowering of the bottle into the neck-down position and again lifting it into the upright position must not be too abrupt and the weight of the bottle holder must be held to a minimum (without sacrificing strength) to minimize vibration. Cast aluminum solved the weight problem and heat treatment answered the strength problem for the bottle holders.

Employs Spring Steel for Wear Surface

To maintain the top grip of the holder in alignment, the top-grip roller pin is provided with a square-shaped extension, which slides in a guiding groove in the lower part of the holder, *Fig.* 3. Since the holder is aluminum, which would not provide a wearing surface to withstand the side thrust and sliding friction of the hardened top-grip roller pin, the groove is lined with a thin flat U-shaped piece of tempered spring steel.

Two deep-groove radial bearings, shielded on one side only and spaced apart about 1½ in., provide the mounting for the lower bottle holder spindle, *Fig.* 3. The bearing end plates are machined with labyrinth rings and packed with grease to protect bearings from moisture.

To reduce weight, the top grip of the holder is also heat-treated cast aluminum. The bottle centering bell is molded Bakelite and mounted in a double-row radial ball bearing shielded on one side only to guard against moisture. The centering bell is machined with labyrinth rings which are packed with grease to prevent entrance of moisture into bearing housing. To prevent scratching of the lacquered bottle caps a medium-soft rubber ring is provided in the bottle centering cup.

DRIVE: Because of limited space, the drive must be as small as possible to deliver the necessary horsepower and yet give a relatively large speed reduction; namely, in the ratio of eighty-eight to one. The solution was the use of a Cone type worm drive. Because of the large number of worm wheel teeth in contact with the Cone worm, substantial reduction in the tooth pitch of the worm wheel is obtained, reducing the size of the combination.

A splashproof brake type motor drives the mixer through a Reeves variable-speed pulley and belt driving a pulley (with fan-shaped ribs) attached to the Cone drive worm, *Fig.* 5. The fanning effect of the driven pulley maintains a circulation of cool air over the drive housing. In event of a bottle jam, the brake motor provides instant stoppage of the mixer.

To maintain a constant bottle spinning speed, irrespective of the number of bottles passed through the mixer, a separate splashproof motor, mounted vertically, is used to drive the spinner wheel. This motor is mounted on the base pedestal and drives the spinner wheel through double-reduction helical gearing. Both the brake and spinner wheel motors are electrically interconnected so that they can be operated from a single pushbutton type of magnetic starter.

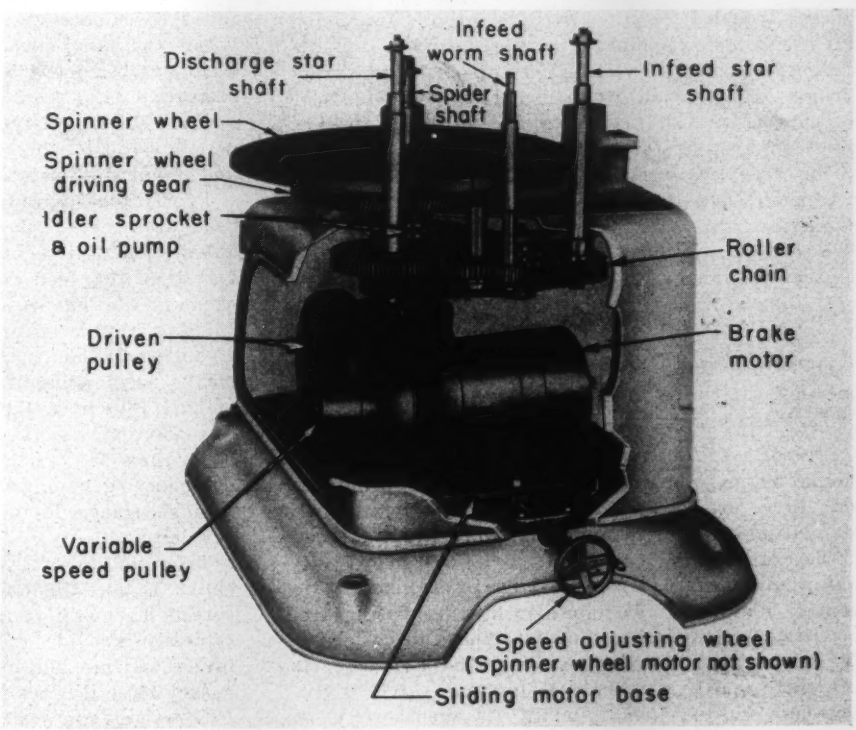


Fig. 4—Left—Infeed and discharge for machine are served by one conveyor. Safety gate protects infeed worm and star. After receiving bottles at left, the holders invert the bottles as they travel around the machine and return them to upright position for discharge

Fig. 5—Right—Main drive for machine spider also drives the infeed and discharge stars and the infeed worm

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One-Way Clutches

... their features and uses with suggestions for indexing and automatic drive applications

ONE-WAY clutches are unusual in the power transmission field because they may be employed in a wide variety of applications. The identical clutch construction might be used, for instance, to backstop a water-pump drive or to feed strip metal to a blanking press. This adaptability makes the one-way clutch a valuable tool to the designer seeking an inexpensive means for automatic control of power transmission.

This article will review applications and discuss details of construction. Before proceeding with the applications, however, it may be helpful to identify the principal types of clutch design. A fundamental distinction may be made between positive and friction types. Included in the positive group are helix clutches, best exemplified by the auto hand crank and ratchet clutches, such as used on clockwork mainsprings. The friction group includes wedge clutches, comparable to the block dragged behind a wagon wheel to keep it from rolling backward; wrapped-spring clutches, resembling a rope snubbed around a capstan; and differential band-brake clutches, in which the band-brake ends are so fastened that one direction of rotation tends to tighten the band, the opposite to loosen it.

Application to Machine Drives

There are three general types of application for one-way clutches: ratcheting or indexing, no-back or backstop and freewheeling drive applications.

INDEXING CLUTCHES: The best-known index is the simple single-pawl ratchet. However, there are many variations used to meet particular problems, such as multipawl for greater adjustability and friction types for infinite adjustability.

An important use for indexing mechanisms is in control equipment. Examples include oscillating arm counting devices, calculating machines, and alternating valves for air cylinders. Since the moving parts are small and light, simple, single-pawl ratchets give low cost and excellent reliability in such service. A similar application is in machine-tool ratchet feeds such as are used on grinding machines and shapers. *Fig.* 1 illustrates the type of special ratchet stop mechanism usually supplied on cylinder grinders.

Intermittent feed devices for strip, ribbon and wire materials constitute still another field of use, such as feed of sheet metal stock through blanking presses or feed of ink ribbon through a typewriter. The one-way clutch is favored because it is simple, reliable and readily varied in stroke; moreover, it acts as a freewheeler to facilitate manual threading or rapid advance.

For feeding strip stock the one-way clutch is usually installed on the wind-up roll, although push-through or both may be used with stiff materials. When the scrap does not emerge in a continuous strip, or when there is no scrap web at all, as in manufacturing nails from wire, push-through feed is required. Nearly all types of one-way clutches may be used for roll feeding, depending on the speed, stroke and required adjustability.

Typical feeds powered by simple connecting link and crank drives are limited to about 90 degrees maximum index. Should it become necessary to secure a greater travel of the feed rolls, intermediate gearing or a rack and gear can be used.

Drive friction on the return stroke or stock weight frequently result in drag-back of the strip, particularly when using a friction wedge-type clutch. A no-back clutch on the other end of the roll shaft is one solution; another is a frictioning device. The latter has the particular merit of also preventing overshooting on high-speed operation and improving feed accuracy. Its disadvantage is that it loads up the drive.

The problem of overshooting is serious when the clutch is used to index heavy machine members, as turrets or chain conveyors. Punch-press dial feeds typically provide a continuous acting metal-disk brake on the dial hub, which for still greater accuracy may also have a wedge-type locking detent.

Overshoot can sometimes be controlled simply with

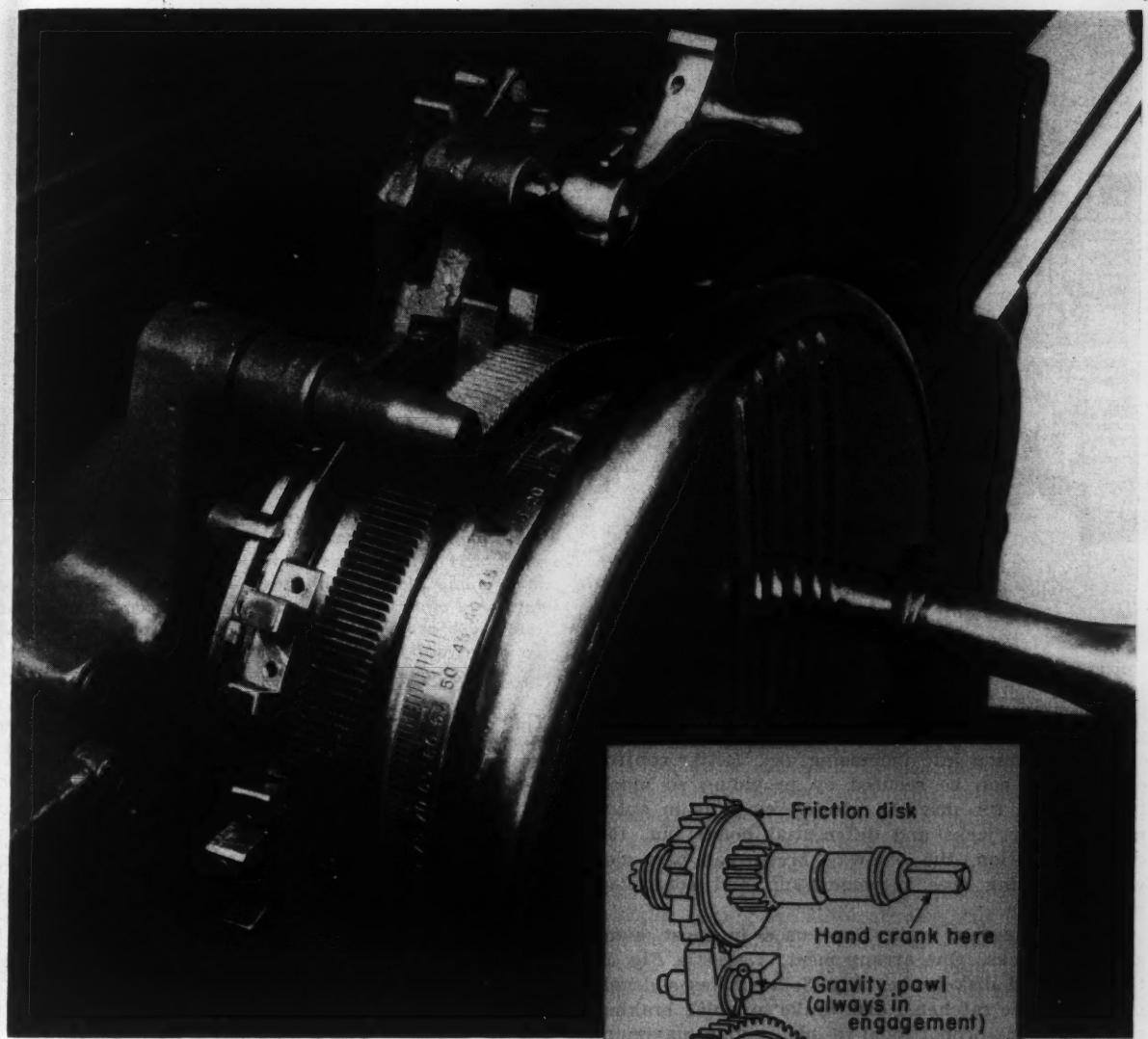


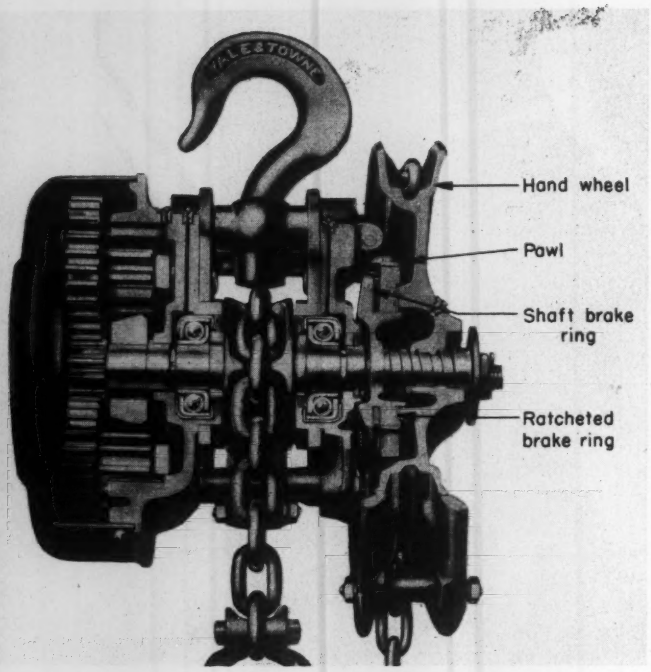
Fig. 1—Top—Ratchet feed mechanism for automatic cylinder grinder, showing adjustable ramp used on ratchet wheel to lift pawl and stop feed at desired dimension

Fig. 2—Above—Simple ratcheted one-way brake used in a hand-operated lift truck to secure controlled lowering and free hoisting

a cam pin to lock the pawl in the ratchet tooth at the completion of index. This is particularly well adapted to drives, where deceleration occurs at the end of the stroke only and there is little chance for the ratchet tooth to coast ahead of the pawl before this time. A construction of wider application is a cam-operated brake to control the dial or turret during deceleration and lock it in position until ready for the next index.

Although outside the scope of this article, it should be noted that the problem of index overtravel can be overcome by going to different (and usually more expensive) indexing means, such as the geneva, the interrupted helical cam, or the cycloidal drive combined with a positive clutch.

BACKSTOP CLUTCHES: The backstop has served from ancient times in the form of the windlass ratchet which, to this day is found in hand-operated hoisting equipment. Such ratchet no-backs do not permit heavy loads to be lowered safely, however, because the pawl must be held out of engagement, with



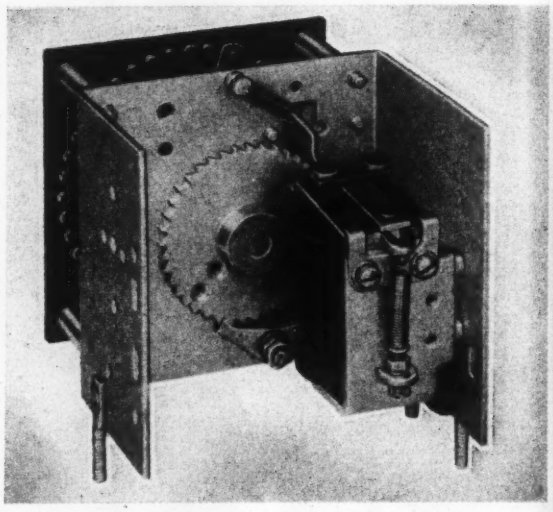


Fig. 3—Left—Mechanism in geared chain hoist, showing easy-operating, ratchet-and-helix one-way brake. Should shaft start to spin under load, it would screw handwheel in to apply proportional pressure on brake ring which is ratcheted against this rotation. To lower load, handwheel releases brake enough to let load follow under control. To raise load, handwheel locks against brake ring and rotates entire ratcheted assembly, similar to Fig. 2

Fig. 4—Above—In ratchet stepping relay, a pawl drives the solenoid armature. A no-back pawl is provided beneath ratchet disk to assure positive action

the possibility that the operator will let the crank get away from him. Other arrangements are accordingly coming into increasing use. Better control of the load can be secured by providing so much friction that the drive is nonreversing, as in automobile screw jacks and differential hoists but, for easier operation, the one-way brake has been developed, giving free movement up and controlled braking down.

Simple one-way brakes are used in most hand stacking trucks. One arrangement is seen in *Fig.* 2, the ratcheted disk brake. *Fig.* 3 illustrates a development of the ratcheted disk brake in which braking action is made proportional to load, giving maximum efficiency under all conditions. Equivalent devices known as "load brakes" are often installed in heavy cranes and derricks to secure full control over the speed of descent.

Backstop clutches are also used on elevating conveyors to prevent a reversal runaway of the loaded buckets in event of power failure. Such protection is essential when the drive is not inherently self-locking. To keep size and cost at a minimum, the backstop is usually installed on a high-speed shaft. Sometimes, however, it is considered more desirable on the free end of the conveyor headshaft to insure against failure of any intermediate member.

Most one-way clutch types can be used successfully for backstop service, although certain manufacturers have brought out specially adapted designs as discussed in a following section. On occasion a conveyor drive must be designed to permit reversal. While this can be done with a manual clutch or brake, the designer should weigh the extra flexibility offered by a solenoid brake in place of the backstop.

The backstop is not only a safety device, it is an integral part of certain mechanisms, e.g. clockwork, which could not function without a no-back on the mainspring. Another example is the ratchet used to prevent reverse running of some self-starting synchronous clock motors. Backstop clutches also insure proper operation of ratchet indexes *Fig.* 4.

FREEWHEELING CLUTCHES: The freewheeling, or overrunning clutch offers innumerable ways of simplifying and improving power transmissions. A simple but essential use of the clutch is in protecting both big and little machines fitted with hand wheels or cranks from inadvertent reversal by the operator. Such clutches are found for instance on business machines, on wrapping machines, and in the control knob of automatic domestic washing machines. In addition to protecting the machine, the clutch also protects the operator because the hand crank does not revolve when the machine is under power.

Acts as a Speed Reducer

Another function of the freewheeling clutch is to convert reciprocation or oscillation into progressive rotary motion, *Figs.* 5 and 6. When driven by a crank on a high-speed shaft, the clutch will operate as efficient speed reducer with potentially a high mechanical advantage. The reduction ratio of the "reducer" may be readily varied by crank throw adjustment and the mechanism is inherently both cheap and sturdy. It is believed that industry might do well to consider making wider use of it for high-ratio speed reduction work when the intermittent output is not objectionable. A few current applica-

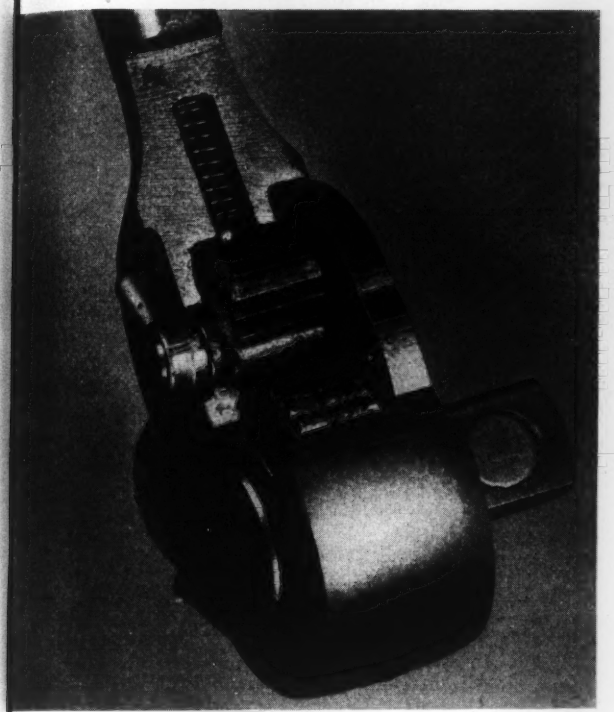


Fig. 5—Ratchet wrench effectively secures progressive rotary motion from a cramped, short-stroke reciprocation. Construction is compact and indicates a method of securing ratchet reversal

tions include automatic lubricating devices, domestic stokers, and certain bulk chemical feeders.

A new transmission, *Fig.* 7, uses multiple cranks to overcome the handicap of intermittent power flow and is an inexpensive unit for variable speed drives. Output speed may be varied from zero to one-quarter of input. It is interesting to note that the same drive principles have been applied for many years on gasoline railroad section cars.

The ratcheting advance is occasionally found in connection with straight-line power sources such as solenoids or fluid-power cylinders. One construction giving virtually continuous advance employs a center-pivot lever, having a pawl at each end oscillated by a double-acting, automatically valved cylinder. Another construction used for cycling employs a double-acting cylinder driving through a rack and gear to carry a machine cam shaft ahead one revolution. A one-way clutch between the gear and cam shaft enables return of the piston for the next cycle.

A related use is exemplified by the lawn mower and the hand-generator flashlight. In these devices a freewheeling clutch affords unrestricted rotation of the driven member after the driving stroke has stopped and vastly increases their effectiveness.

Often, clutch coasting action is employed to advantage where the driven member has high inertia and where sudden stopping of the drive would result in damage. In an electric blasting machine, for instance, parts would be damaged if a clutch did not permit the rapidly spinning armature to coast ahead after

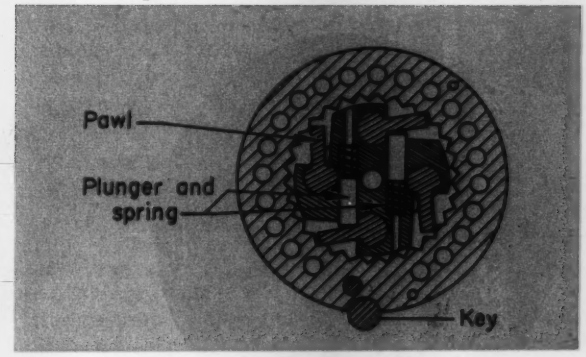


Fig. 6—A section through the helix-ratchet rotation mechanism of a pneumatic rock drill. The internal pawl assembly tapers into a rifle bar (not shown) which fits rifled hole in piston. In action, each forward stroke of piston rotates rifle bar and pawl assembly one tooth counterclockwise. On each return stroke, piston in turn is forced by pawls to advance, also counterclockwise

the rack reaches the bottom of the stroke. The freewheeling clutch has been found desirable also in high-speed drives such as gear-driven superchargers, to eliminate fatigue-inducing stress reversals. Another illustration is the clutch used in certain aircraft jet engines to prevent undersirable reactions between compressor and turbine.

A somewhat different application of the overrunning clutch is on electric starting motors for internal combustion engines. These intermittent-duty motors must be disengaged after a start to prevent destructive overspeeding and power-wasting drag on the engine. The overrunning clutch performs this function simply and automatically, *Fig.* 8. Incidentally, the automotive application presents a difficult design problem because of the high pinion speeds (to 40,000 rpm), the low internal friction required,

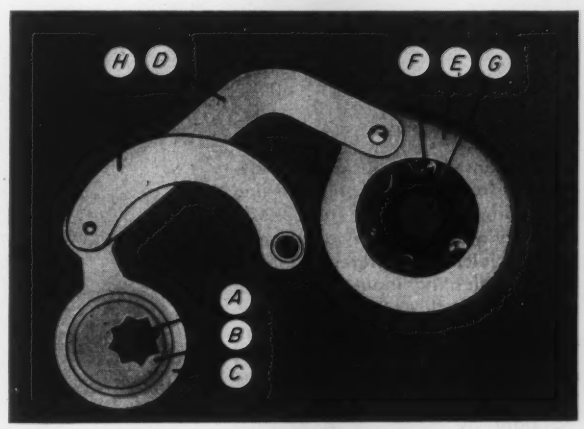


Fig. 7—Mechanism for a variable-speed transmission. Eccentric B is driven at constant motor speed and, through arm D, oscillates housing E of a wedging-roller ratcheting clutch mounted on the output shaft. The stroke, and hence speed ratio, can be varied by shifting the free end of control link H so as to change direction of throw of C. Several elements are mounted together to smooth output

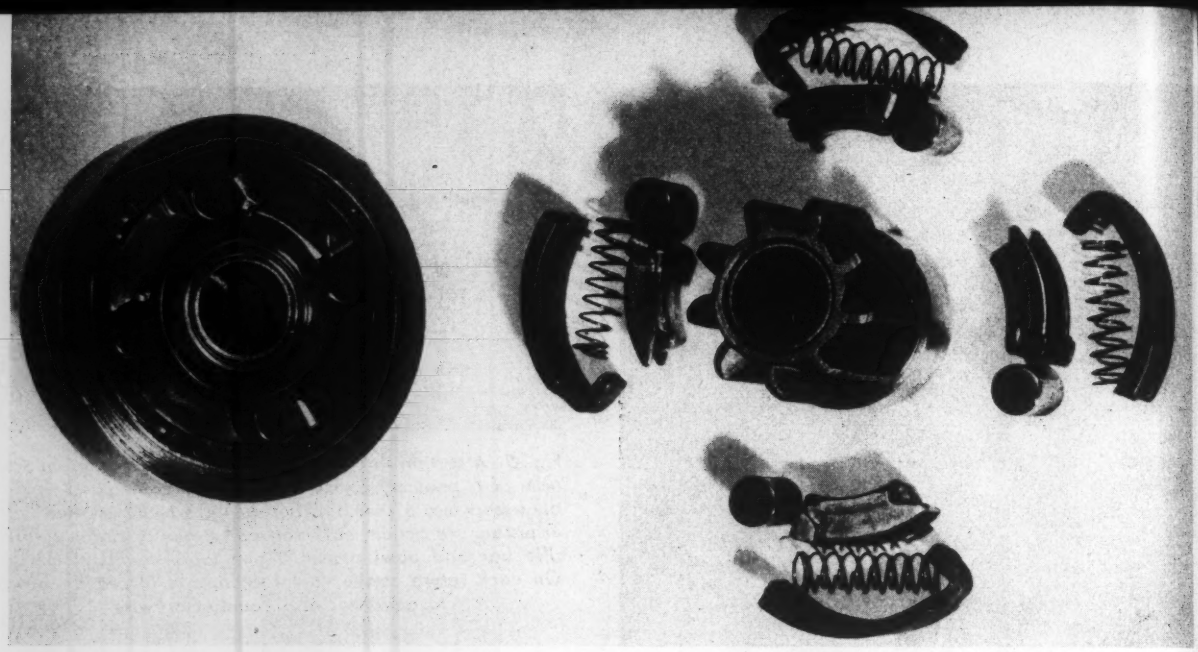


Fig. 8—Above—Parts and assembly of ingeniously-designed wedging-roller overrunning clutch used in automotive starting motors. A 2½-degree roller locking angle is used on the cam pieces. Clutch will transmit 120 lb-ft torque

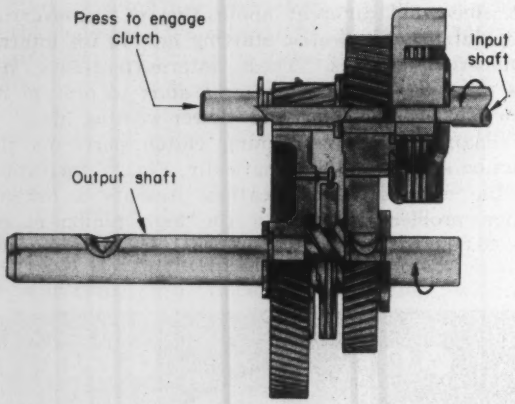


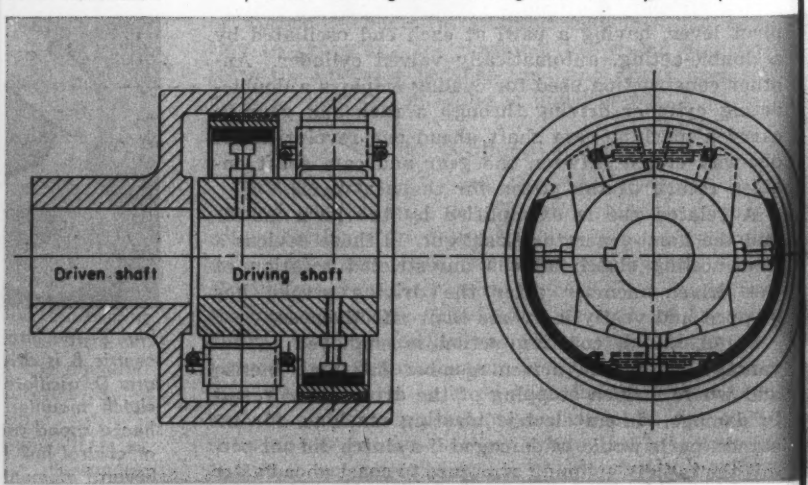
Fig. 9—Above — Two-speed helix-type transmission, originally designed for an automatic washing machine, now is used for general applications. For high-speed operation, clutch is engaged by pressure on floating input-shaft extension; power flows from large driving gear to smaller driven gear. Meanwhile, the bull gear is idling on output shaft and is urging the triple-threaded control nut lightly to right by means of a retarder spring sliding in groove around nut. For low-speed operation clutch is released and, as output shaft slows below bull gear speed, the self-tightening retarder spring carries control nut to left until it strikes driving lugs on bull gear and locks

and the range of operating temperatures (−20 to 150 F). In addition, the clutch must be low-cost and must operate, without relubrication, for the life of the car.

Small starting motors, operating through large reductions, have been used successfully with overrunning clutches to "break away" exceedingly heavy machines. The main motor automatically takes up the drive after friction has been reduced to the normal running condition. This arrangement also performs as an inching drive and is used in newspaper printing presses and sheet and strip steel mills as a means for threading-up.

The same arrangement has been used to make a two-speed drive, wherein a fan or other equipment is direct-connected to a high-speed motor which is in turn coupled to a smaller geared motor through a

Fig. 10—Below—This reversible slip ring centrifugal starter is provided with retention springs on the friction bands to prevent rubbing until driving shaft is at preset speed



one-way coupling. The principal advantages over a multispeed motor are the complete flexibility in speed ratio obtainable and the improved motor efficiency on low-speed, light-load operation. Operation is also smooth with such a two-motor arrangement.

A two-speed drive can be secured with a single-speed motor with gear reduction, a friction clutch, and a one-way clutch, *Fig.* 9. The particular advantage of this arrangement over a standard clutch and gear-shift drive is that at no time is the motor completely disconnected from the load. It is interesting to note that the basic principle is used in automatic automobile transmissions. Another variation has recently appeared on the market, an automatic transmission containing two overrunning clutches so arranged that by reversing the motor, two speeds are provided, one at direct motor speed and the second at a preselected reduction without changing the direction of rotation of the output shaft.

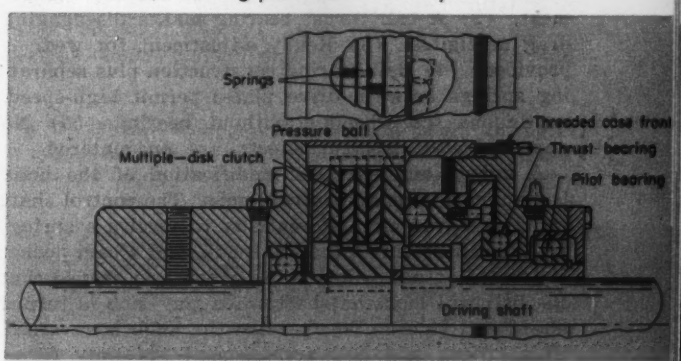
One of the best-known applications for the freewheeling clutch is the standby, or dual drive. For instance, an important circulating pump, normally driven by an electric motor, is coupled through a freewheeling clutch to a steam turbine to insure against electric power failure. It is customary also to use this dual drive on boiler feed pumps and furnace draft fans.

It is common to see two internal combustion engines in a dual drive for one generator. This arrangement facilitates maintenance and permits running both engines together on peak loads, thus minimizing the size of engine required. The advantage of the freewheeling clutch over the manual cut-off clutch is that it is fully automatic. The disadvantage is that auxiliary starting means are required. When selecting the freewheeling clutch, it may be desirable to use a type that will permit starting the engine unloaded whether or not the other prime mover is running. No standard one-way clutch will answer this need, but certain centrifugal starters can be converted by the addition of springs to prevent rubbing contact while freewheeling as shown in *Fig.* 10.

POSITIVE DRIVES: One of the simplest of one-way clutches is the pin and helical slot arrangement which will be recalled in connection with the auto hand crank. A compression spring is sometimes added to assist disengagement, and instead of a helical slot, a triple-threaded screw may be used for control, often to shift a pinion in and out of engagement with a driven bull gear. Several variations are illustrated in *Figs.* 3, 9 and 11.

In the overrunning clutch, *Fig.* 12, the helix is modified into a cammed ball race. This design offers several operating advantages: (1) Driving torque and shocks are taken by oil-cushioned, sintered-bronze clutch plates and not by the cams and balls,

Fig. 12—Below—Overrunning clutch uses cam-actuated balls to transmit working pressure to a multiple disk clutch



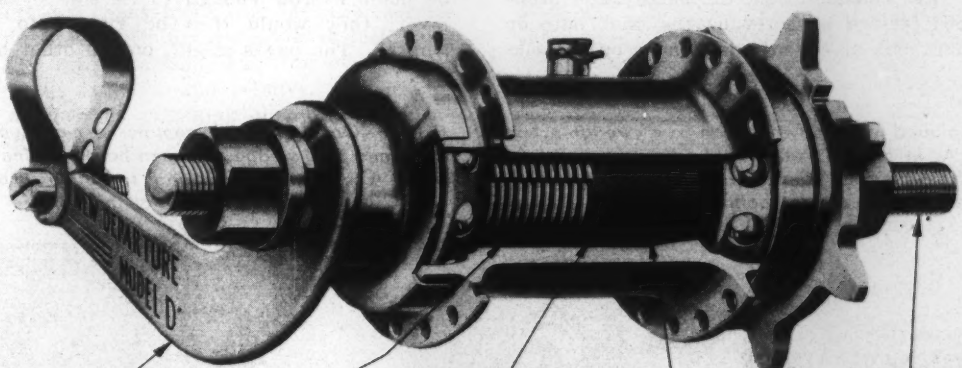


Fig. 11—Above—Bicycle coaster brake mechanism has an internal-threaded control sleeve guided by a retarder spring (tail showing in slot). Control sleeve engages a threaded extension of sprocket. When sprocket drives wheel, sleeve is drawn right until it locks in wheel-hub taper. Grooves on sleeve cut through oil and prevent slippage. When rider stops pedalling, relative motion between sleeve and sprocket forces sleeve to left, disengaging from wheel hub, permitting coasting. When pedals are backed, sleeve is carried farther to left until it engages teeth on sliding washer keyed to axle. Pressure on the disks, keyed alternately, applies the brake

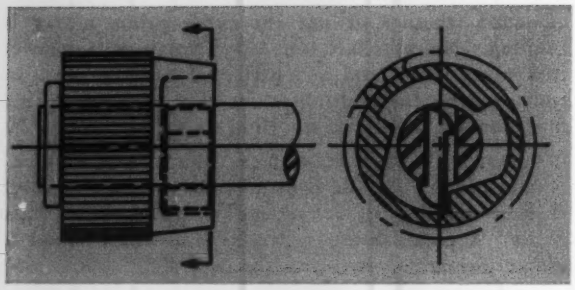


Fig. 13—Above—Section through a springless ratchet lawnmower clutch. When gear is driven in freewheeling direction, the alternating ratchet slopes carry the shuttle pawl back and forth freely; when gear is reversed, pawl locks on nearest tooth. A hook under pawl head engages shaft to insure positive locking

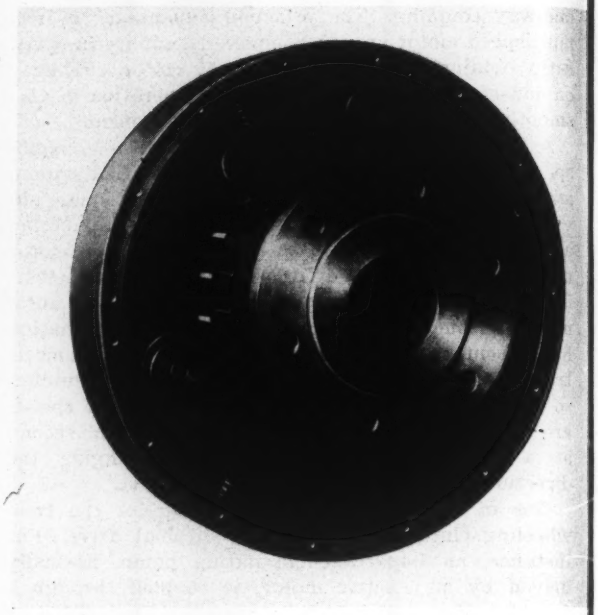


Fig. 14—Right—Ratchet backstop clutch uses centrifugal force to prevent contact and wear of pawls while freewheeling. Dust cover has been removed

avoiding cam brinelling and giving smooth engagement. (2) A ball thrust bearing makes disengaging drag negligible. (3) Ready adjustment for wear is provided. (4) Ball-bearing construction plus separating springs between drive plates permit high-speed, continuous free-wheeling without heating. (5) No appreciable centrifugal effects are encountered.

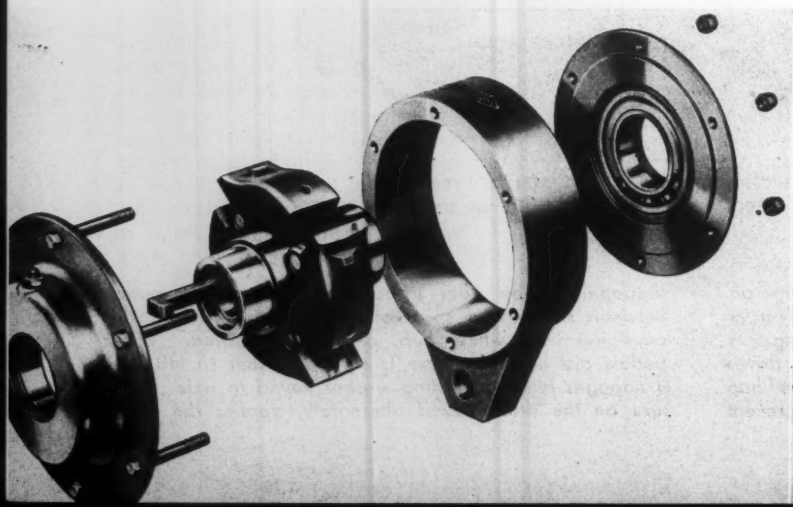
A simple and effective modification of the helix is employed in small control knobs. The control shaft end is grooved circumferentially to receive a spring-loaded ball detent in the knob hub, and a cam pocket is formed at one point of the groove, so shaped that the ball will lock the shaft when the knob is turned ahead and ride up freely when the knob is reversed. This groove and ball arrangement not only forms a one-way clutch with a fixed engaging relationship but also holds the knob on the shaft. Where cost is the first consideration, the small knob problem has been tackled by screwing the knob onto or into the shaft end, as the winding knobs on a clock.

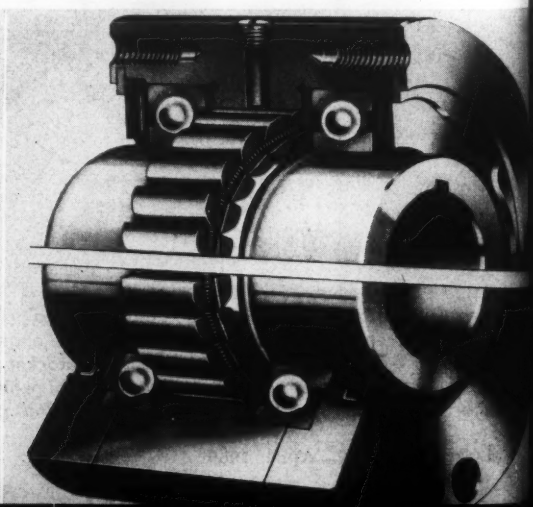
The ordinary ratchet construction is widely used because it combines simplicity, positive action, and long life. In contrast to the helical slot and pin, it permits engagement at many intermediate angular positions. It has, however, several drawbacks that limit its utility and have led to design modifications. One is the fact that throw adjustment is ordinarily limited to increments of the tooth spacing.

Finer adjustments can be secured with multiple pawls at vernier spacing so that only one pawl drives on any given stroke. With five pawls, for instance, a ratchet with 1/16-inch pitch teeth can be adjusted for a 1/80-inch change of advance when driven by a suitable adjustable throw crank. These pawls may be made narrow enough to fit side by side on one pivot; they would then be ground to progressive lengths. The pawls might, on the other hand, be in-

Fig. 15—Exploded view of a friction-pawl wedge-action backstop. As in Fig. 14, centrifugal force prevents rubbing while freewheeling. Clutch is lubricated and sealed. Anchoring hole is at bottom of housing

Fig. 16—One-way clutch contains a maximum number of hardened steel wedges between hardened inner and outer races, with dual garter springs to obtain simultaneous engagement of all members





stalled on individual studs spaced at progressive distances around an oscillating ring, thus permitting each pawl to span the full width of the ratchet tooth. Another method of securing fine ratchet adjustment is to adopt friction pawls, similar to those in *Fig.* 15.

In *Fig.* 13 is illustrated a compact, economical construction used on lawn mowers to obtain positive pawl action and assure proper operation despite dirt or neglect. This mechanism would obviously be unsatisfactory in larger sizes or at high speeds because of pawl hammer. Accordingly, in ratchet clutches designed for high-speed operation, backstops, springs and centrifugal force are utilized to keep the pawls out of engagement while freewheeling, *Fig.* 14. In this design, at least one of the multiple pawls would come into engagement by gravity even if all springs were broken.

FRICTION DRIVE: Friction is applied in various ways in a number of different clutches to secure one-way action in addition to other desirable properties. The outstanding characteristic of a friction drive is smooth, shockless engagement. The principal limitation is variability of action, depending on the condition and contact of the rubbing surfaces. Contact pressures should be watched, as in many clutches there is a real danger of surface damage by brinelling if overloaded.

In *Fig.* 15 is a backstop clutch similar in many ways to *Fig.* 14, particularly in the centrifugal unloading. The important difference is the use of a friction wedge action rather than teeth to lock. The friction pawls lock almost instantly at any rotational position and there is no danger that associated equipment will be subjected to impact when the pawls take hold.

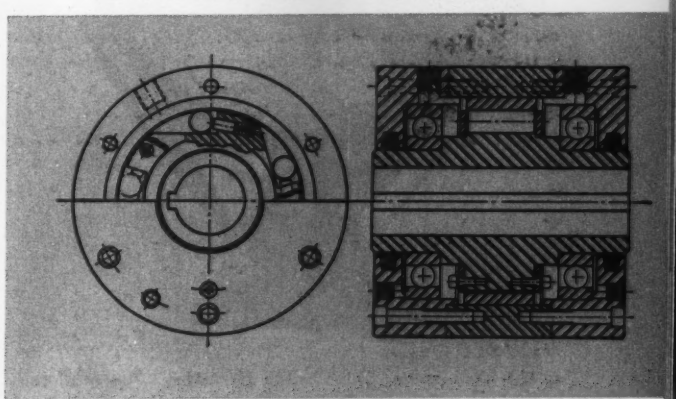
Similar metal-to-metal wedge action is found in the sprag clutch, *Fig.* 16, named after the wooden sprags placed under the rear wheels of wagons to keep them from rolling back downhill. This type clutch may be used for backstop service where it gives a shockless engagement, for indexing service where it offers an infinite degree of adjustability with virtually zero backlash, and for overrunning service where it is quiet and reliable. Because of the large number of engaging members and the absence of any localized action of the races, the clutch has an unusually high torque capacity for a given size and weight. An oil film formed while freewheeling minimizes wear and drag.

A somewhat different clutch made, with variations, by a number of manufacturers is the cammed wedging roller type, *Figs.* 8 and 17. Instead of shaped sprags, rollers are provided in cam pockets and individually spring loaded. This clutch is used successfully for all three types of service, although the type of service should be specified because heavier springs are supplied for ratcheting.

Two other all-metal friction designs are illustrated in *Figs.* 18 and 19. *Fig.* 18 shows an inexpensive wedging roll-feed with the special merits of reversibility and nonslip action regardless of the presence of oil or grease. *Fig.* 19 illustrates a low-cost design based on the time-honored principle of the expanding or contracting wrapped spring, comparable in action to a rope wrapped around a capstan. Partly because of the characteristic cushioned engagement, wrapped spring clutches have been used successfully as heavy-duty load-brakes on cranes, as well as on the light-duty applications illustrated.

A third basic friction design is the differential band brake, shown in *Fig.* 20. Many other one-way clutch designs could undoubtedly be worked out beyond

Fig. 17—Below—Wedging-roller one-way clutch comprises a cylindrical outer race, and side cage plates for rollers. Individual spring plungers urge rollers lightly against outer race, assuring simultaneous gripping



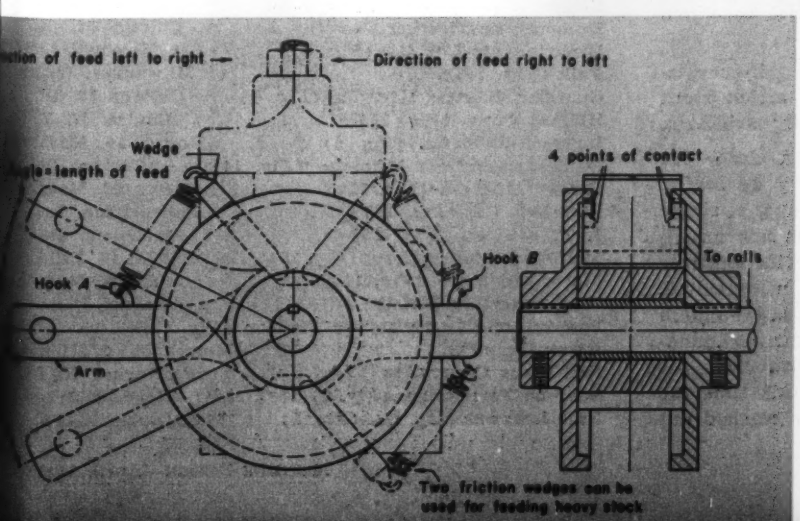


Fig. 18—Left—In this punch press strip-feed clutch, a double-notched wedge member fits between facing flanges of two driven disks and rests in a cam notch in driving arm. To feed, arm is driven clockwise, which cams up lower end of wedge so that the notches cramp on disk flanges and entire assembly turns with arm. On reverse stroke, cam pressure is relieved and spring slides wedge around flanges

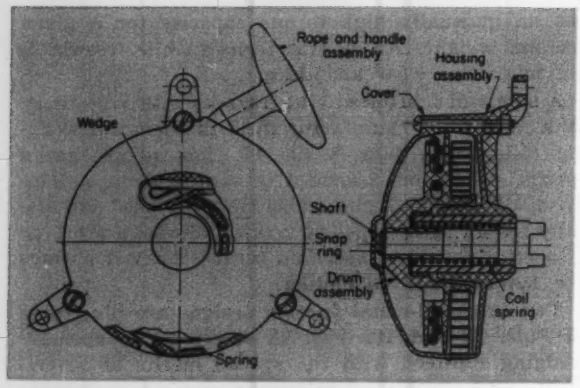


Fig. 19—Above—Built-in pull-rope starter and wrapped spring clutch used in a portable gasoline chain saw. Normally in light contact with starter shaft, the wrapped spring winds up and grips shaft when rope is pulled, releases as soon as shaft overruns

Fig. 20—Differential band-brake backstop. Powerful self-energizing action results from the unequal-arm bell-crank in the base, to which each end of the band is connected

those described, such as variations of the self-energizing automotive brake or various wedge-action block brakes. However, none are commercially available.

SELECTION: As a guide to selecting a clutch for a specific application, the following may be useful.

General: Inadequate clutch size is the largest single source of trouble. If adverse conditions are anticipated such as rapid pick-up of high-inertia loads or rapid cycling, the manufacturer should be consulted. Best design under any conditions calls for the clutch to be accessible for preventive maintenance examination. Space should be provided for unbolting the housing and for sliding parts out. At this time the slimy gray wear sludge, if any, may be washed out with solvent and the clutch relubricated, otherwise slippage in metallic friction types may result.

The effect of possible hot, dirty, or wet surrounding conditions must be considered, and consultation with the manufacturer regarding seals is suggested when such conditions are anticipated.

Special Constructions: Most clutches can be obtained with built-in brake drums, with sprockets, pulleys or gears, or with flexible coupling action; or parts may be bought separately for incorporation in other assemblies. On large orders the manufacturer will usually design special variations as needed; for instance, long, small-diameter designs or short, large-diameter designs for limited spaces, built-in backstops for indexing drives, and stamped or die-cast constructions for minimum-cost fractional-horsepower drives.

Speed of Operation: Antifriction bearings are desirable in any clutch operating at standard motor speeds. In high-speed freewheeling applications, roller clutches should be run with the outer member overrunning the inner, sprag clutches just the oppcsite. This will normally prevent overheating.

May Require Supplemental Brake

Engaging Action and Direction: Certain clutches have a tendency to lock slightly when driving and may need a back torque of several per cent to force release, depending on condition. This is normally unimportant for backstop and freewheeling applications, but on accurate indexing drives, a supplemental brake or no-back is usually required.

With the exception of the helix types, direction of action in most clutches can be reversed by transposing parts. Reversal by the flick of a lever can be built into some, especially ratchet designs.

Freewheeling Drag: In clutches where the friction members stay in contact, such as sprag, roller, or wrapped spring types, values for freewheeling drag are given by the manufacturers as ranging from 0.5 per cent down to 0.1 per cent of full-load torque rating, depending on design and size.

The author acknowledges the helpful co-operation of the following companies for furnishing illustrations for this article:

Cincinnati Grinders Inc. (*Fig.* 1) ..... Cincinnati 9, O.
Eclipse Machine Division (*Fig.* 9) ..... Elmira, N. Y.
Economy Engineering Co. (*Fig.* 2) ..... Chicago 24, Ill.
Electric Auto-Lite Co. (*Fig.* 8) ..... Toledo 1, O.
Falk Corp. (*Fig.* 15) ..... Milwaukee, Wis.
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Ingersoll-Rand Co. (*Fig.* 6) ..... Boston 16, Mass.
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Kinney Mfg. Co. (*Fig.* 12) ..... Boston 30, Mass.
Link-Belt Co. (*Fig.* 20) ..... Philadelphia, Pa.
J. P. Madden (*Fig.* 10) ..... Bethlehem, Pa.
McCulloch Motors Corp. (*Fig.* 19) .. Los Angeles 45, Calif.
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Revco Inc. (*Fig.* 7) ..... Minneapolis 2, Minn.
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Snap-on Tools Corp. (*Fig.* 5) ..... Kenosha, Wis.
Yale & Towne Mfg. Co. (*Fig.* 3) .... Philadelphia 15, Pa.

Hydraulic Differential Drive

. . . provides infinitely variable speed ratios for airplane cabin pressurization

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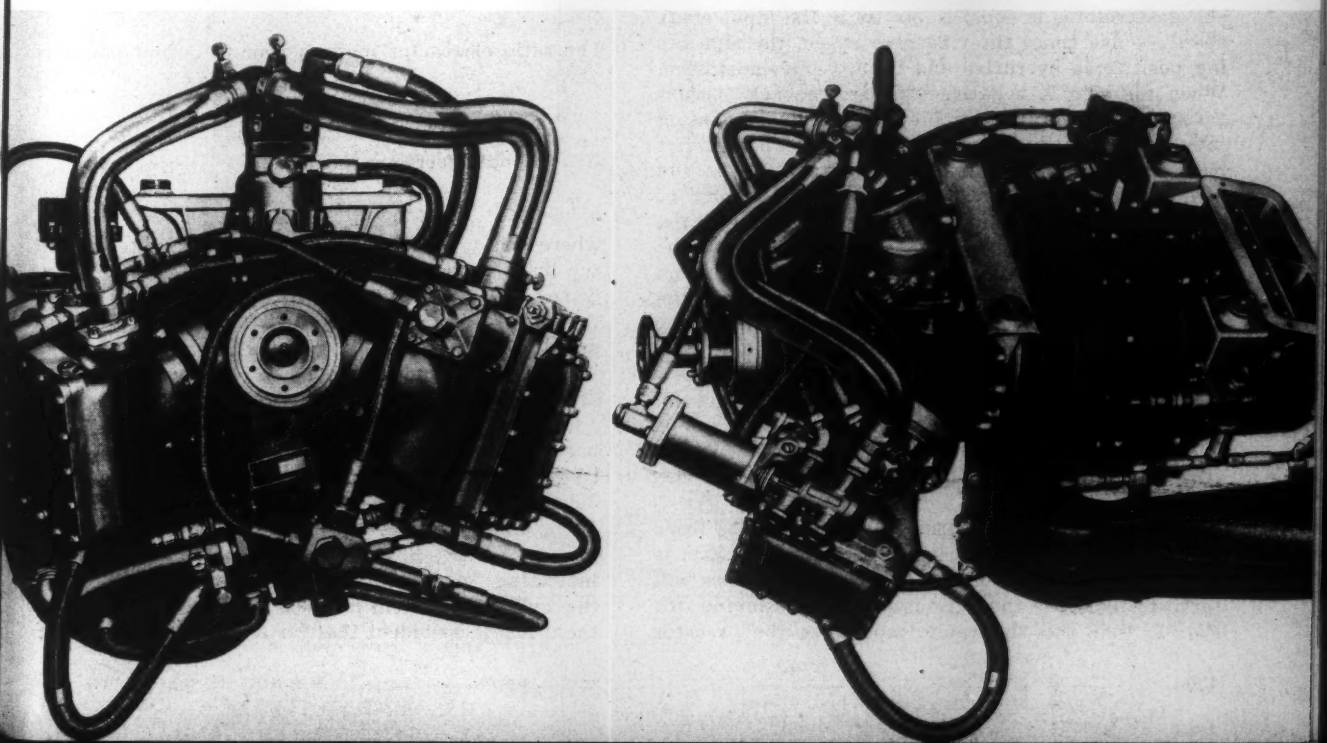
COMBINING the differential gearing principle with a variable-displacement hydraulic circuit for ratio control provides a unique positive, infinitely variable type of transmission. Such a system is being used for the impeller drive of airplane cabin superchargers on Lockheed Constellations. While the method of speed control is not new—having been known for many years in the rail car field as the "Waterbury" drive—the transmission as designed for aircraft use involves numerous features of timely interest to designers, particularly in view of the attention now being focused on all forms of automatic transmissions. Developed by AiResearch engineers the pressurization compressor complete with its variable-ratio drive, *Fig.* 1, is capable of maintaining a substantially constant pressure of air under varying engine speeds and flight conditions. High efficiency and freedom from surge and pulsation are achieved through the compensating features of the differential gearing of the transmission and the variable-displacement hydraulic circuit.

It can be seen from the illustration that the variable-ratio subassembly of the unit, *Fig.* 1, bolts directly to the compressor which is of the two-stage type with backward curved impellers. Driving connection between the transmission and the compressor is by means of a slender quill shaft, *Fig.* 2. The Vickers hydraulic equipment consisting of one fixed-displacement motorpump and two variable-displacement motorpumps, is mounted on the gearbox.

To compensate for the varying speeds of the input drive shaft, which is direct-connected to the accessory case of one of the main engines, the transmission operates with a step-up ratio capable of varying be-

Fig. 1—Front and side views of variable-ratio supercharger drive utilizing combination of hydraulic motorpumps and a planetary gear set



tween 3.4 to 1 and 9.7 to 1. Actual input speeds may be as high as 8000 rpm, with full power transmitted at speeds down to 3300 rpm. Normal speed of the output shaft is 32,500 rpm, this being maintained automatically through the combination of the hydraulic pumps, hydraulic motor, and planetary gear set, *Fig.* 3.

Ratio Control Initiated At Venturi

SEQUENCE OF OPERATIONS: Automatic control of the speed ratio is initiated at the venturi located in the discharge elbow of the air compressor, *Fig.* 3. Operation of the compressor control valve shown on the circuit diagram is dependent upon the flow velocity and hence on the velocity of the impellers. The control valve determines the oil pressure acting upon the servo cylinders which in turn control the variable-displacement hydraulic pumps. When the output velocity is low the compressor control valve assumes the position shown, the pumps are shifted toward "HI", and the pressure from the pumps energizes the hydraulic motor, which rotates the planetary ring gear to effect a speed-up of the sun gear driving the impellers. This condition, known as "overdrive", is brought about smoothly whenever the impeller velocity is lowered, the same as when the engine speed is reduced.

At higher engine speeds the tendency is toward increased impeller velocity. When this happens the compressor control valve stem moves up, permitting the oil acting on the servo cylinders to be by-passed to the oil tank, thus relieving the pressure. During this "underdrive" condition the servo cylinders adjust the pumps to "LOW" and the hydraulic motor, now driven through the planetary transmission, serves as a pump (and the variable displacement units as motors) automatically bringing about a reduction in the speed of the sun gear.

On this installation the actual speed at which the sun gear rotates is equal to six times the input shaft speed $\pm$ five times the ring gear speed, the sign being positive in overdrive and negative in underdrive. When the sign is negative, "power feedback" occurs through the hydraulic circuit and the power passing into the planetary gears through the planet carrier becomes greater than the power output of the sun gear. Similarly, in overdrive the power output of the sun gear is greater than the power input to the planet gears, the difference being made up by the hydraulic circuit. The greater the deviation from the nominal 6 to 1 planetary ratio the greater is the relative amount of power handled by the hydraulic circuit. Power losses are determined by hydraulic circuit and gearing efficiencies, this drive having better than 85 per cent overall efficiency over its normal operating range.

Basic Theory Relatively Simple

BASIC THEORY OF DIFFERENTIAL DRIVE: The basic theory of a differential drive of the type discussed is relatively simple. A brief mathematical analysis will serve to illustrate the various points. Assuming, for *Fig.* 3, that the "hydraulic motor" is the "reactor motor" let:

R_o = Basic planetary speed ratio (output/input) with reactor member locked
R_1 = Gear ratio (output/reactor motor) with input shaft locked
R_n = Net gear ratio (output/input) with reactor turning
T_o = Output torque (lb-in)
T_1 = Reactor motor torque
T_2 = Input shaft torque
N_o = Output rpm
N_1 = Reactor motor rpm
N_2 = Input shaft rpm
P_o = Output power (hp)
P_1 = Reactor motor power
P_2 = Input shaft power

For any differential drive:

$$R_n N_2 = N_o$$

$$N_1 = N_o \left[\frac{R_n - R_o}{R_1 R} \right]$$

$$= N_2 \left[\frac{R_n - R_o}{R_1} \right] \quad \ldots\ldots (1)$$

If frictional effects are neglected

$$T_1 = T_o R_1 \quad \ldots\ldots (2)$$

Also:

$$T_1 = T_2 \frac{R_1}{R_n}$$

from which it follows that

$$T_2 = T_o R_n$$

The power transmitted is

$$P_o = P_2 = \frac{T_o N_o}{63000}$$

$$P_1 = \frac{T_1 N_1}{63000}$$

The ratio of reactor motor power to output power is

$$\frac{P_1}{P_o} = 1 - \frac{R_o}{R_n} \quad \ldots\ldots (3)$$

If frictional effects are included:

$$T_1' = T_1 (1 \pm f) \quad \ldots\ldots (4)$$

where f is a nearly constant factor characteristic of the drive. The sign is positive for $R_o/R_n < 1$ (overdrive) and negative for $R_o/R_n > 1$ (underdrive). For the drive described in this article $f = 0.03$, approximately.

It might be added that $P_2' = P_o/\eta$ where η is the drive efficiency.

From the foregoing equations certain factors of importance appear. As Equation 3 indicates, the relative amount of power handled by the reactor motor is determined by the factor $1 - (R_o/R_n)$. Thus, for $R_o = 6$ and $R_n = 9.7$, $P_1/P_o = 38.2$ per cent and for $R_n = 3.4$, $P_1 P_o = -76.5$ per cent the negative sign indicating "pumping" instead of "motoring". Since the hydraulic system is much the heavier portion of the drive, it is evident that for $R_o/R_n > 2$, the differen-

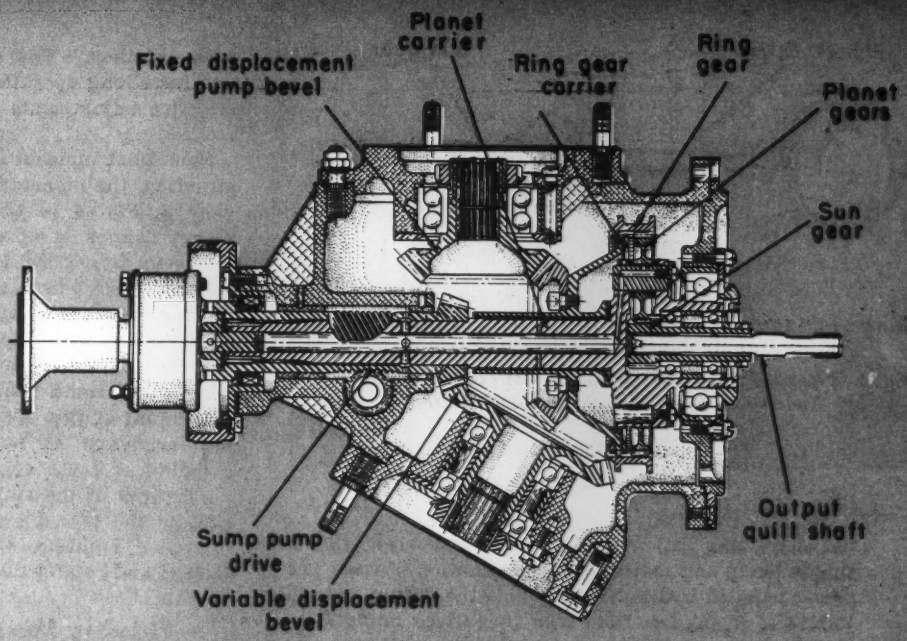
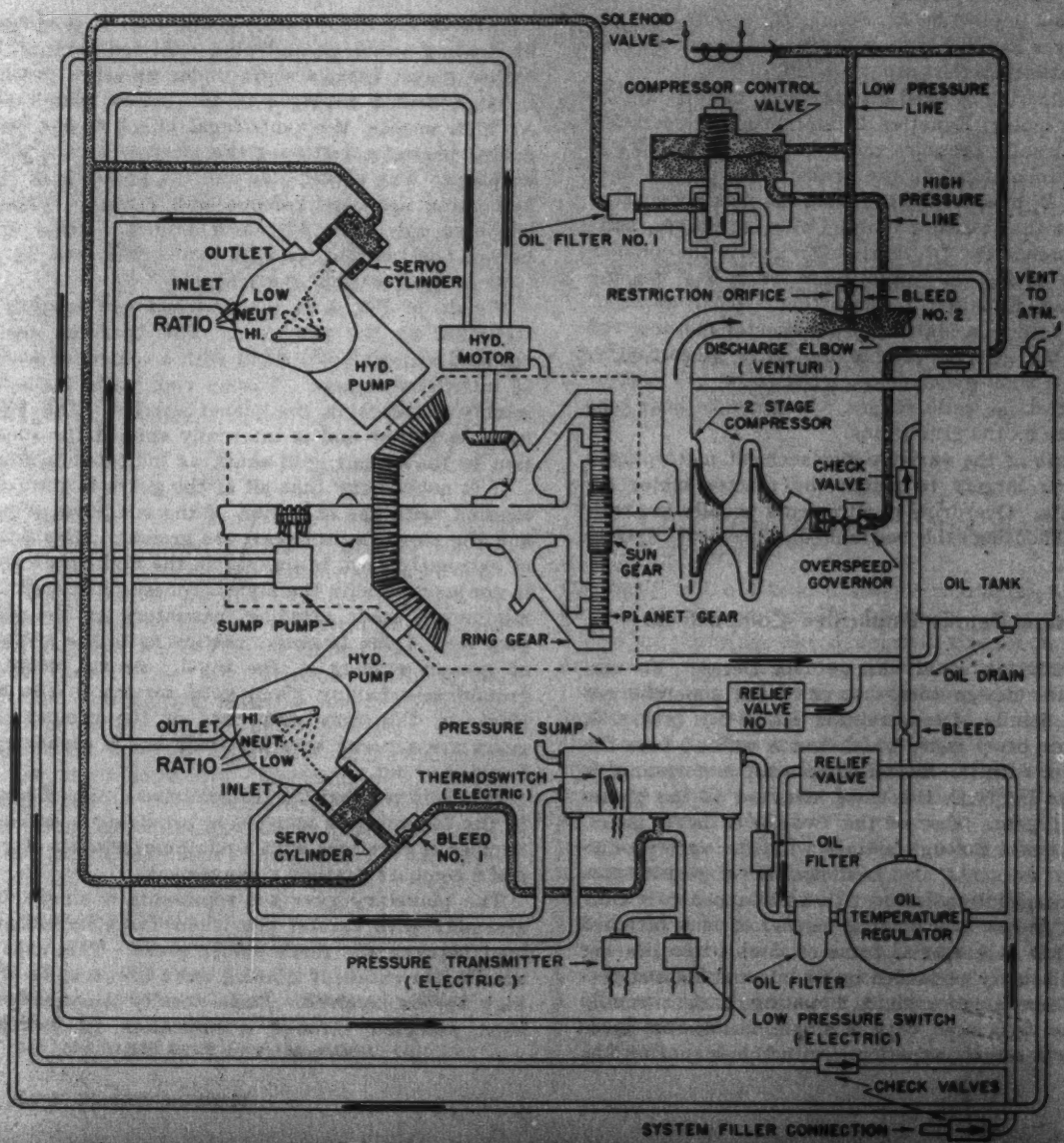


Fig. 2 — Cross section of gearbox showing planetary transmission, at right, and bevel gearing for connection with fixed and variable-displacement motorpumps

Fig. 3 — Hydraulic circuit and control system shown for high ratio operation with hydraulic motor driving ring gear of planetary transmission. For low ratio operation the hydraulic motor serves as a pump and the variable-displacement hydraulic pumps as motors



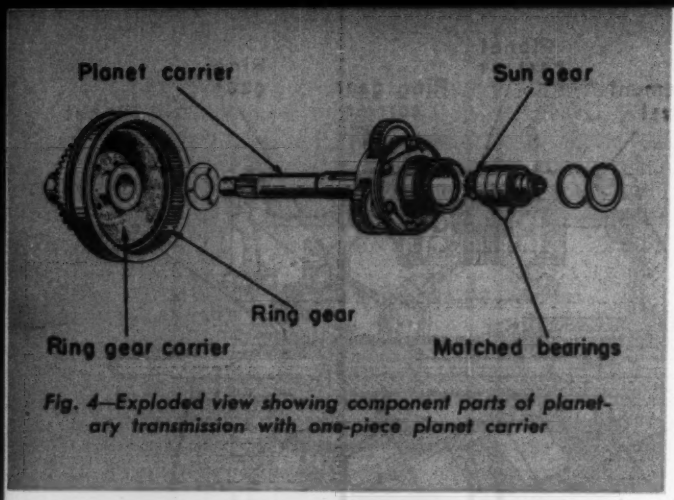


Fig. 4—Exploded view showing component parts of planetary transmission with one-piece planet carrier

tial drive has little to offer when compared with a simple pump and motor drive combination because the power handled hydraulically equals the output power! Values of R_n/R_o less than zero (negative) imply a reversal of output rotation as compared with the output rotation implied by R_o. For $R_n/R_o = 0$ or $R_n/R_o <$ 0, the power handled hydraulically also becomes equal to or greater than the output power.

In considering a fixed-displacement motorpump as a reactor motor, Equation 2 is of prime importance. Since hydraulic pressure must be limited, and T_1 is directly proportional to the pressure for a given motorpump, R_1 must be set to match the maximum value of T_o and the pressure limits (with, of course, suitable allowance for friction). Obviously, the smaller the value of R_1, the lower the value of T_1. On the other hand, Equation 1 shows that a small value of R_1 tends to give a high value for reactor motor speed (N_1) at the extreme ranges of R_o/R_n. Accordingly, the torque and speed characteristics of the reactor motor affect the ratio ranges. In general, these considerations fix the gear ratios.

The sizes of the variable-displacement motorpumps are chosen largely to match the reactor motor requirements. Overdrive requirements usually are critical in choosing the variable-displacement system elements.

Design Details Emphasize Compactness

CONSTRUCTION FEATURES OF THE DRIVE: To emphasize the design compactness of the supercharger gearbox, details of construction are shown in *Fig.* 2. The upper bevel gear, which has a splined bore for connection with the fixed-displacement motorpump, is shown mating with the bevel attached to the planetary ring gear. One of the two lower bevel gears, which connect through splines with the variable-displacement elements, also is shown. Bevel gear mountings are conventional except that the upper gear runs in a double-row ball bearing instead of in a matched pair. This is a special form of double-row bearing which is heavily preloaded by its internal fits and provides a much more rigid mounting than normally would be associated with a single double-row bearing. Field service experience with this mounting has been excellent. The gears are ground spiral bevels and have long operating life. Shim or thrust washer selective adjustments are provided for all bevel gear meshes.

Somewhat unusual in its carrier and ring gear construction, the planetary gear set is shown in exploded view in *Fig.* 4 to illustrate the arrangement. The planet carrier is of one-piece construction, and the sun gear is carried in ball bearings which pilot inside the carrier to minimize concentricity errors. Held in place by internal snap rings, the floating ring gear slips onto a spline carrier. The splines are merely extensions of the internal gear teeth and are a loose fit to allow small radial movements of the ring gear. The intent of this design is twofold: To insure gearing accuracy by making provision for internal grinding of tooth contours; and to permit the small movements of the ring gear to equalize load distribution on the planet gears. To minimize concentricity tolerance build-up, the ring gear spline carrier is bushed and rotates directly on the planet gear carrier.

Grinding Mandatory for Planet Gears

Each of the three planet gears, *Fig.* 4, is mounted on floating bushings and pins. Oil fed through the hollow planet carrier shaft under pressure proceeds through drilled passages to the hollow planet pins. At high speeds, the centrifugal effect of the radial drilled passages increases the oil flow to the planet bushings. The planet pins and the planet gear teeth and bores are casehardened and ground. Floating bushings are of a high-lead bronze. Their split-halves construction permits center oil feed to the inner and outer rubbing surfaces.

Visible in *Fig.* 4 is the matched ball bearing installation on the sun gear. The bearings are assembled without preload but with a controlled amount of internal looseness. A snap ring holds the subassembly in place in the planet carrier. The pinion itself is hollow and is internally splined for connection to the output quill shaft, as indicated in *Fig.* 2.

It is noteworthy that all of the gears in the transmission, with the exception of the sump pump gears and the plastic spiral gear, are ground. Maintenance of extremely close tolerances on the high-speed gears, in conjunction with the high hardness (Rockwell C-60 minimum), made grinding mandatory on the planetary set. There is some question as to the necessity of ground gearing on the bevel gear set, some unground sets having given good service in the field. However, the wear properties of the ground bevel gears are superior whether their use is economically justified or not.

It is believed that the transmission design discussed in the foregoing is unusual in providing a maximum number of functions with a minimum number of gears and a compact gearbox arrangement.

The planetary gear set represents a simple basic assembly with careful provisions for equalization of load between the three planet gears. This care has resulted in excellent gearing wear life in spite of the high speeds involved. Some penalty necessarily has been accepted through complication of individual

(Concluded on Page 236)

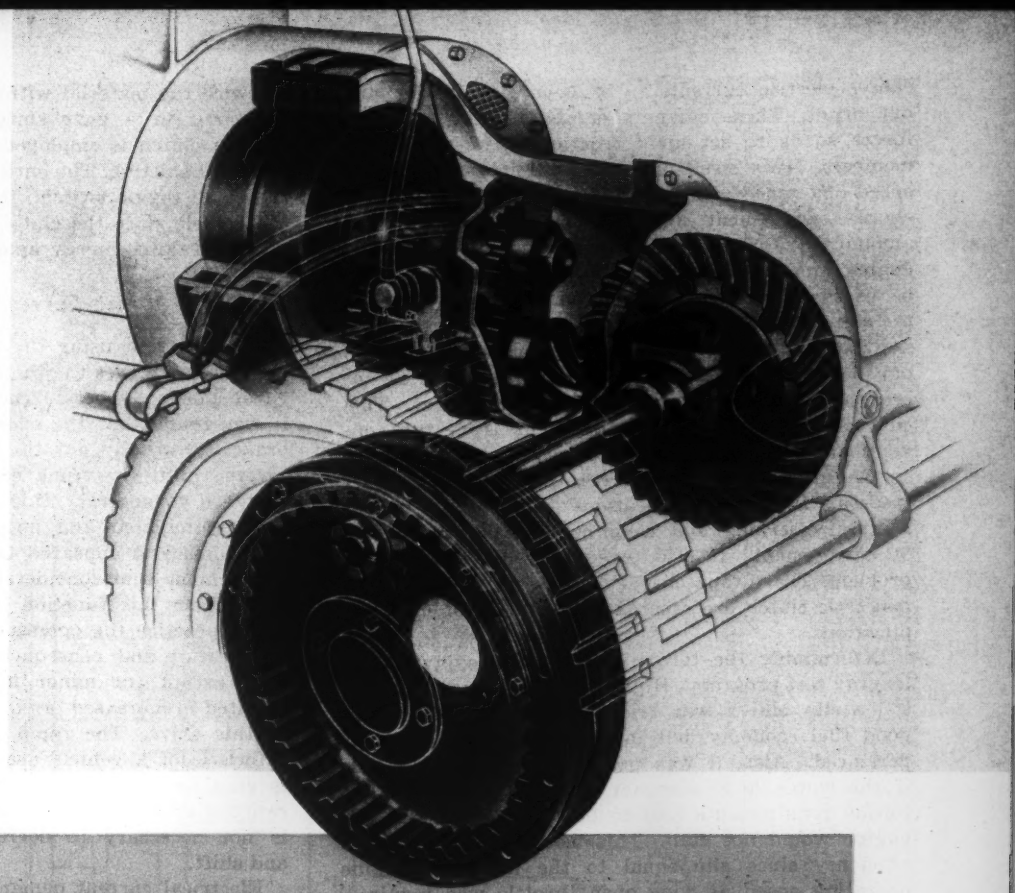


Fig. 1—Cutaway view of transmission which obviates gear shifting for forward and reverse operation of industrial truck

By R. L. Jaeschke
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Novel Transmission

...IMPROVES TRUCK PERFORMANCE

BECAUSE of the large number of clutching operations required and the degree of control necessary, engine-driven industrial trucks have always required undue maintenance. By necessity the vehicle is compact so that it can operate in confined quarters, complicating clutch design and maintenance problems. Actual cost of a friction clutch of the conventional automotive type is small but repair and replacement are difficult and expensive.

Demands for a better clutch mechanism indicated that it should require little maintenance and have control characteristics not found in the friction clutch. Also, it should not necessitate redesign of the entire truck to meet dimension requirements. An ideal clutch should have the following characteristics.

1. No wearing parts
2. Low intertia in the output member
3. Rapid response to the operator
4. Smooth control and good inching characteristics
5. Size no larger than presently used clutches
6. Little or no auxiliary control mechanism
7. Reasonable cost.

Obviously, all of these desirable characteristics are difficult, if not impossible, to meet.

Several years ago the dynamatic type of magnetic drive was proposed for truck clutches. At that time this clutch was too large and too heavy for suitable application. Electrical current demands were high and the cost was excessive. Design of a practical clutch, however, was achieved through the use of better materials and new pole shapes. The final design as shown in *Fig.* 1 is the result of extensive tests and experiments and is a revolutionary transmission employing two clutches.

Each clutch consists of two members. One, having a coil and suitable magnetic structure, replaces the engine flywheel and is somewhat heavier than the conventional flywheel. The second or output member is a light drum capable of dissipating heat caused by slippage in the clutch.

When energized, the coil in the magnetic structure

causes electric currents to be generated in the output drum. These currents create counter magnetic forces so as to set up a reaction between the two members. This mechanical reaction increases as the speed difference (slip) increases, resulting in smooth clutch engagement and eliminating a substantial amount of torsional vibration transmitted by the engine since a small amount of slip must always be present to cause transmission of torque through the clutch.

Illustrated in *Fig.* 2 are torque curves. As the slip or speed difference between the two members increases this curve shows an increase in torque. This characteristic is somewhat different from the conventional friction clutch, the torque of which increases as the speed difference between the two members decreases. In many applications this results in unnecessary jerking and roughness during the period of engagement. It also presents a difficult control problem. The torque characteristics of the dynamatic magnetic clutch are repetitive and exact for most applications.

Determining the torque characteristics involved a lengthy test program. High efficiency being desirable, a low-slip clutch was tried first. This resulted in good fuel economy, but increased tire wear was experienced. Also, it was possible to stall the engine if the clutch lever was not operated carefully. The torque requirements had to be changed so that the engine would not stall. This meant a slight loss in efficiency since slip equal to the minimum engine operating speed at wide open throttle was required, approximately 200 rpm in this case.

Peak torque available at high slip speed must be below the torque required to slip the wheels, yet give good acceleration. Therefore, the torque curve should be substantially flat, falling off at lower speeds to conform somewhat to the engine torque curves.

Characteristics Are a Compromise

For these reasons, the clutch torque curve results in a compromise between a high-efficiency clutch and one that would have the desired characteristics. The actual difference in gasoline economy, compared to a conventional transmission, is hardly noticeable although there is a measurable difference. However, improved handling qualities of the truck more than pay for the slight loss in gasoline economy. In many cases it is possible to improve the productive output of the vehicle due to the rapid clutching characteristics of the system. By varying the materials, the pole shapes, and drum structure, it was possible to obtain the torques shown in *Fig.* 2. These follow the engine curve satisfactorily.

Two magnetic structures and drums are shown in *Fig.* 3. The two clutches (forward and reverse) are connected to a constant-mesh gear system, as illustrated in the cutaway view, *Fig* 1, so that forward and reverse shifting is unnecessary. A toggle switch mounted on the steering column is used for changing direction. It was found that the ease of changing direction through the use of a simple toggle switch resulted in considerable improvement in work time. It was possible on some applications to move as much as twice the material with a truck of this type when compared to a gear shifting mechanism. When a friction clutch is employed, considerable time is used in actual shifting. The entire process is now done with the small toggle switch. It is not necessary to brake the vehicle since the clutch will actually absorb most of the braking energy against the engine.

Also Serves as a Brake

Possibility of using the reverse clutch for braking purposes increases the thermal capacity requirements. After placing several trucks in factories for tests, it was found that the operator seldom used his foot brake, he usually put the direction toggle switch in reverse until the truck came to standstill and then switched to neutral. This use of the clutch had not been anticipated and no provision had been made for it, since it appeared to the engineers developing this scheme that considerable skill would be required to perform this function satisfactorily. After a few hours practice the operators become acquainted with this feature and, consequently, the foot brake is little used except for minor holding requirements. This resulted in increased brake life, an unexpected feature of this drive. The rapid forward and reverse characteristic of a vehicle used at factories where space is at a premium or for moving loads in and out of railroad cars, results in considerable time saving. It is not necessary to decrease engine speed, clutch, and shift.

Electrical current demand for this 180 lb-ft clutch is approximately 30 amperes at 6 volts or 180 watts. These current demands on a 6-volt system may seem high but, compared with industrial applications, they are extremely low. Further research is now being conducted to decrease this value. Torque requirements, materials used in the magnetic structure, and air gap dictate the excitation needed. The air gap between the clutch members is 0.020 to 0.028-in. and is determined by the clearance in the engine main bearing and manufacturing tolerances. A 50-amp continuous-duty generator connected to a conventional heavy-duty battery is used for the electrical current supply.

The electrical control system is simple, consisting of only the 3-position switch, (forward, neutral, reverse) and a rheostat. For most inching operations, the switch can be employed. Occasionally, however, very slow and gentle movements are required. So that increased slip can be obtained for such operations, the rheostat provides lower current flow through the clutch. This control is used where extremely heavy loads are to be moved and where the vehicle speed is below that chosen through the gear system and minimum engine speed.

The rheostat controller consists of an eight-step contact making mechanism connected in series with the clutch coil. Electric dissipation is low, being only 45 watts maximum. A conventional voltage-regulator system is used for generator control.

In eliminating metal to metal contact between driving and driven members, wear and tear on the clutch is kept to a minimum. Maintenance thus far has indicated that only the brushes require replacement.

Fig. 2 — Right—Torque characteristics of clutch plotted against slip for various coil voltages

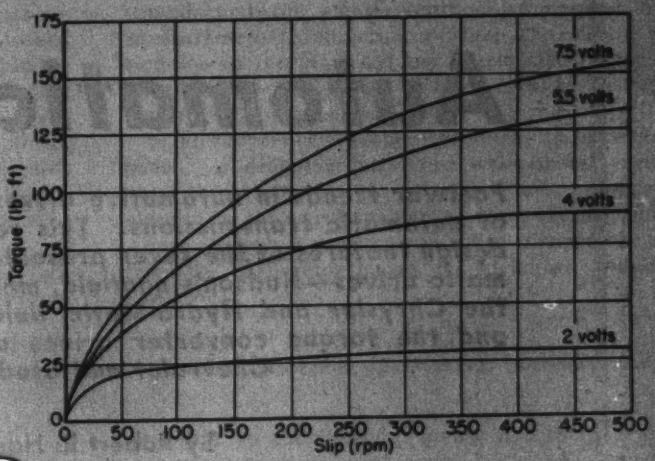
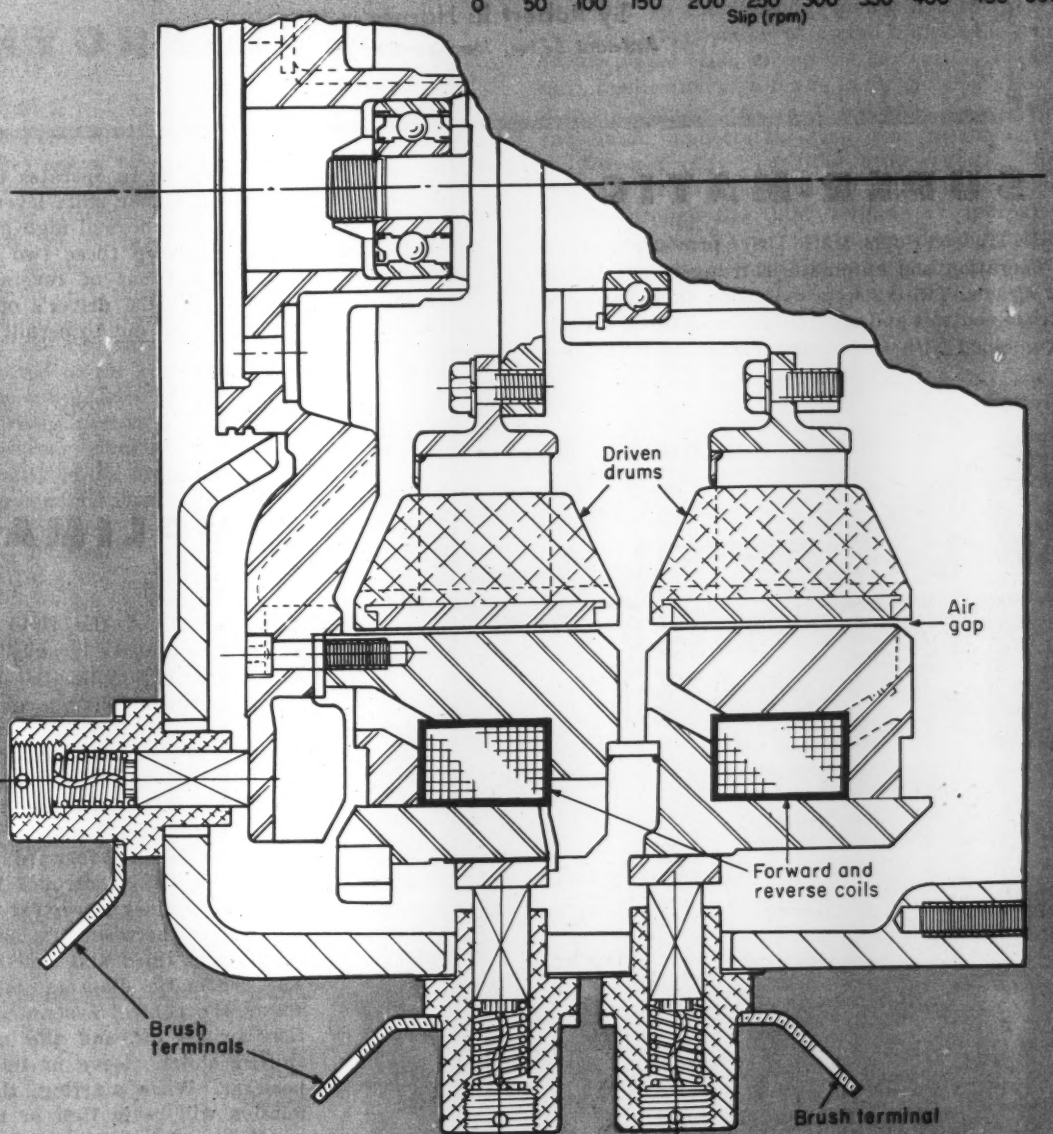


Fig. 3 — Below — Detail of clutch elements. Coils are mounted on flange coupled to engine shaft. Driven drum on right side is bolted to a hollow shaft. Through its center left drum drives a solid shaft



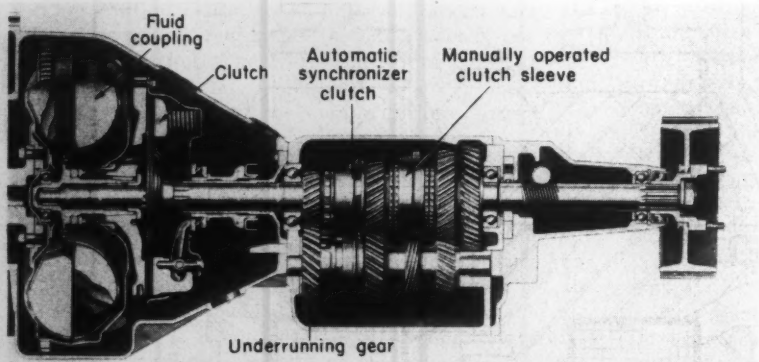
Automatic Drives

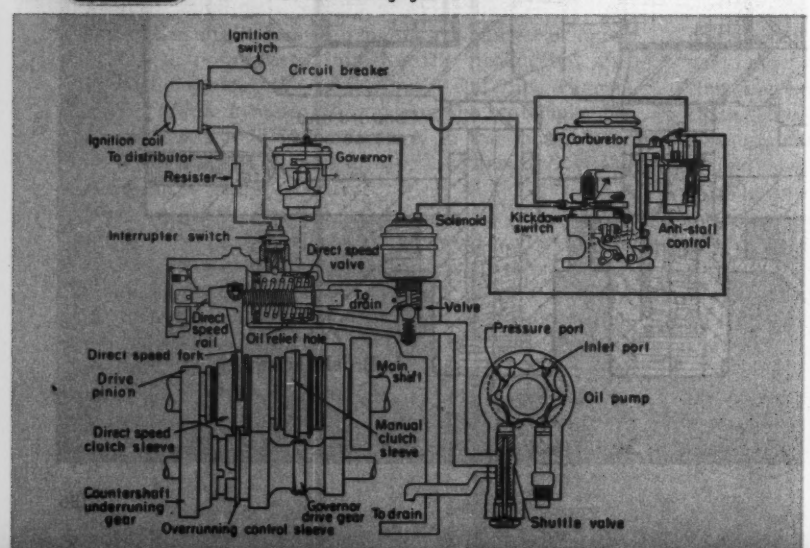
Postwar trends in automotive drives have produced a flood of automatic transmissions. This review outlines the major design features of the seven present-day passenger car automatic drives—Hudson's nonfluid, mechanically-operated unit, the Chrysler and Hydra-Matic fluid-coupling transmissions, and the torque converter drives used by Buick, Packard, Chevrolet and Studebaker.

By Robert R. Harnar
Assistant Editor, Machine Design

SUPER-MATIC

THE Hudson Super-Matic Drive provides overdrive operation and automatic shifting in normal forward speeds. Two vacuum cylinders, their associated electrical controls and appropriate mechanical linkages are combined with a standard transmission and overdrive unit. One of the cylinders operates the clutch and the other actuates the gear shift lever to shift from the "pick-up" gear into normal high gear. Both up and down shifting between these two speeds is automatic. Shifting in and out of reverse or low gear is always manual. At the driver's option, the automatic controls can be cut out to permit standard shifting.





SIMPLIMATIC

SIMPLEST of the fluid coupling type automatic transmissions is the Chrysler Simplimatic, left, consisting of the coupling, a conventional clutch and a semiautomatic four-speed transmission. Reverse, first and second, or third and fourth ranges are manually selected, shifting being automatic between the two speeds in either forward range.

An electrically-controlled hydraulic cylinder provides power for the automatic shifts between first and second speeds and third and fourth speeds. The schematic diagram, below left, shows the control system with ignition switch off and the manually-shifted clutch sleeve in the neutral position. When starting, the transmission will be in first or third, depending on the position of the manual control lever. As the car accelerates, governor weights driven from the output shaft move out and, at 12 mph

(assuming the car is being started in third), the governor points are opened. Opening these points de-energizes the solenoid circuit, allowing a spring-loaded valve to open and admit oil pressure to the shifting cylinder. Here the piston compresses an engaging spring which tends to move a synchro sleeve into engagement with a spline attached to the drive pinion. However, an upshift will not occur until the accelerator is momentarily released, decreasing the torque transmitted and permitting movement of the sliding synchronizer clutch.

When car speed drops below 10 mph (when in high range), governor points close, the solenoid is energized, valve is pushed down, the direct speed piston vented, and the engine ignition interrupted by a pressure-actuated interrupter switch. Cutting the ignition for an instant eases the transmitted torque and allows the spring-loaded shifter fork to disengage synchro sleeve and pinion.

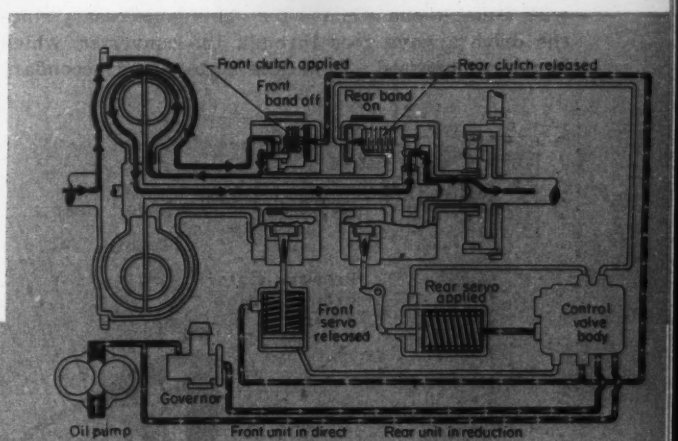
A kickdown switch operated from the accelerator pedal, for dropping into third when more power is needed, is located on the carburetor and grounds the circuits in the same fashion as the governor points. Above 42 mph, a vacuum-controlled plunger operated from the carburetor makes the kickdown switch ineffective.

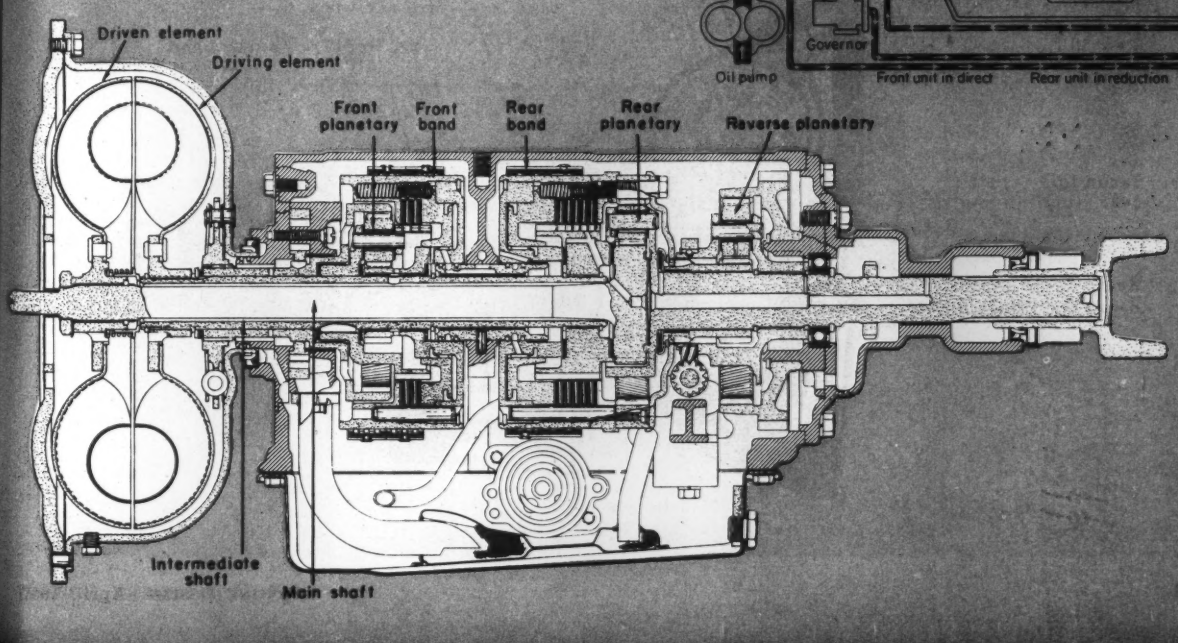
HYDRA-MATIC

ANOTHER fluid-coupling, automatic transmission combination, General Motors' Hydra-Matic, below, makes use of two two-speed forward planetaries and one reverse planetary in series, together with a hydraulic control system, to give four speeds forward and one in reverse. Power is applied direct from the engine to the ring gear of the front unit. In this front unit, the output is taken from the planet cage, which is in turn connected direct to the impeller of the coupling and—through a clutch—may be connected to the ring gear of the rear planetary unit. The sun gear of the rear planetary is connected to the driven element of the fluid coupling. Output from the rear planetary is taken from the planet cage directly to the propeller shaft.

Proper combinations of brake band and clutch application give four possible forward speeds, one of which is illustrated schematically at the right.

In the front unit, application of a brake locks the sun gear to the housing, giving a 1.45 to 1 speed reduction; application of a clutch locks the sun gear to the planet cage, giving a 1 to 1 ratio. In the rear unit, application of a brake locks the ring gear to the housing, giving 2.63 to 1 reduction; application of the clutch connects the ring gear to the output of the forward unit. Under the latter condition the ring gear is driven mechanically while the sun gear is driven through the fluid coupling at approximately the same speed, 62 per cent of the torque being transmitted mechanically and 38 per cent through the fluid





coupling. This differential drive is used in third and fourth gears, resulting in reduced slippage in the coupling and better economy at ordinary driving speeds.

Clutches and bands are applied by oil pressure or springs. Governor-controlled valves direct oil pressure to the proper pistons in response to car speed, depression of the accelerator pedal and setting of the manual shift lever. Normally, all up and down shifting is fully automatic with a forced downshift from fourth to third available by pushing the accelerator past the full throttle position.

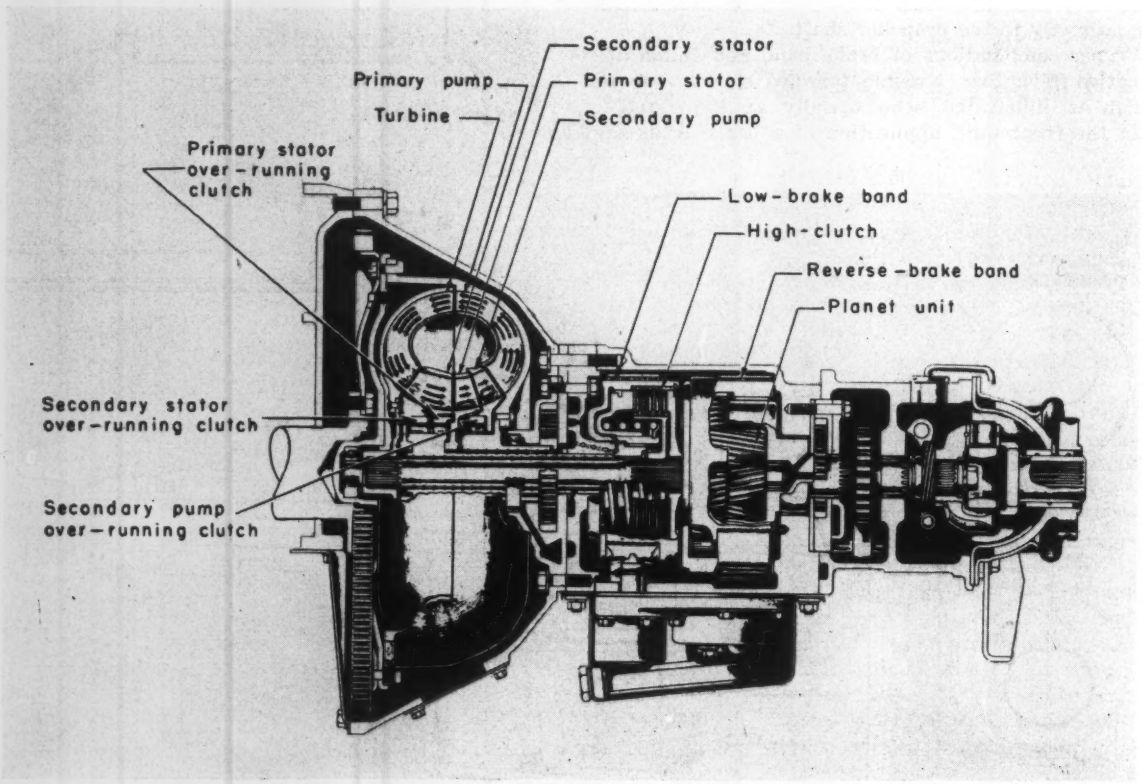
Reverse is obtained through a third planetary set. Because of the high reaction torque in reverse, a toothed arm is employed instead of a brake drum to lock the ring gear. This arm is mechanically actuated from the control lever under the steering wheel.

DYNAFLOW

FIRST of the torque converter type automatic transmissions, the Dynaflow (MACHINE DESIGN, March, 1948), below, is a refinement of the unit developed by Buick for tank destroyers during the war. It is probably the simplest of present-day converter drives, consisting essentially of a five element polyphase converter (aluminum plaster-mold castings), a planetary gearset and hydraulic controls. Maximum torque multiplication in the converter is 2.25.

Buick's four-phase converter is designed to give greater efficiencies at high speeds than is possible with a single-phase converter. This is necessary since the drive always goes through the converter, which is never bypassed. Dynaflow stators and secondary pump freewheel out of the way at speeds that would otherwise produce shock losses and decreased efficiency.

A planetary transmission is employed to provide emergency low and reverse speeds. The planetary unit has two sun gears. The rear, or driving sun gear is splined to the input shaft; the forward, or reaction, sun gear is secured to one element of a friction clutch through which it can be locked to the input shaft for 1 to 1 ratio. This forward sun gear is also secured to a brake drum through which it can be locked to the housing for 1.82 to 1 emergency low-speed reduction. The ring gear is integral with another brake drum through which it can be locked to the housing for 1.82 to 1 reverse drive. Two sets of planet pinions are secured in the carrier which delivers the output: the reverse pinions engage the ring gear and the reaction sun gear; the low pinions, which are of double length, engage the driving sun gear and the reverse planet pinion.

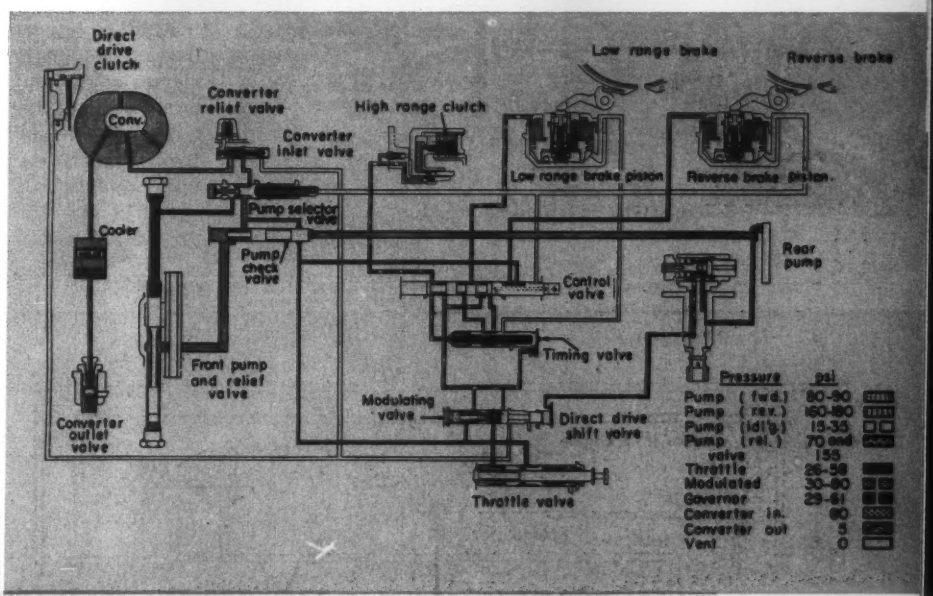


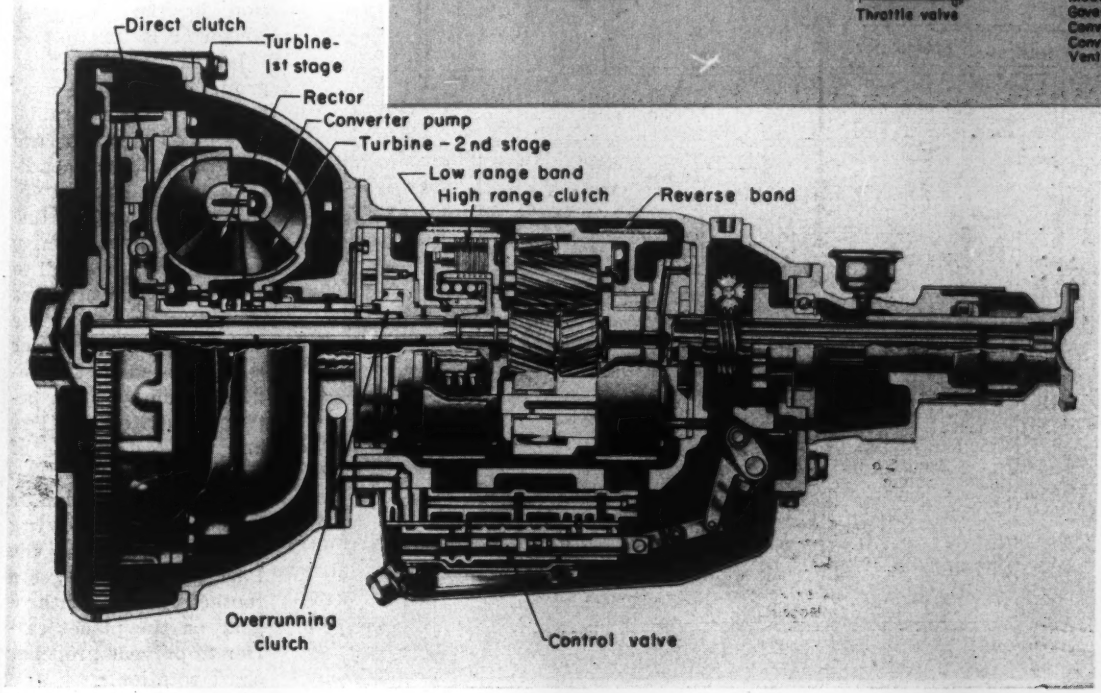
ULTRAMATIC

PACKARD'S contribution to the growing number of automatic passenger car drives is the Ultramatic (MACHINE DESIGN, July, 1949), below. Here again, a torque converter coupled with a planetary gearset and appropriate hydraulic controls provides high, low and reverse speed ranges. In this case, a three-element converter using cast aluminum alloy parts is employed—a pump driven by the crankshaft, first and second stage turbines bolted together and rotating as a unit, and a single reaction member equipped with a one-way sprag type overrunning clutch.

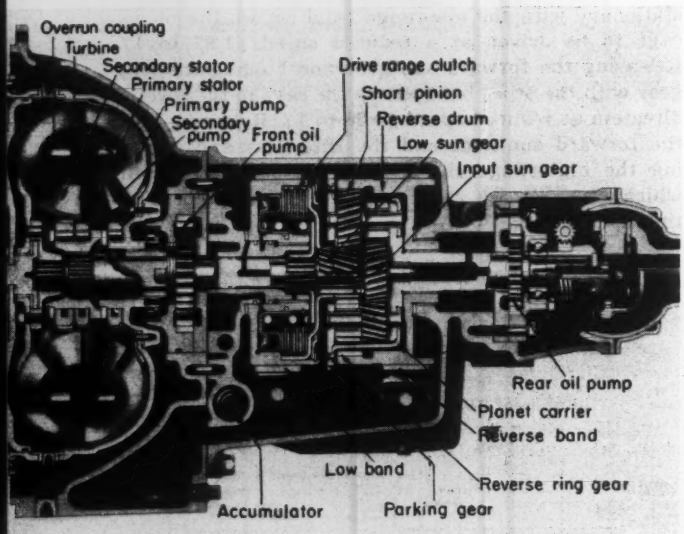
To obviate slippage losses in the converter when it is acting as an ordinary fluid coupling (i.e., at 1 to 1 torque ratio), the Ultramatic utilizes an automatic friction clutch between the crankshaft and the main drive shaft to provide a direct mechanical drive, bypassing the converter. This provision permits toleration of the drop in converter efficiency which occurs at turbine speeds over 1600 rpm. Two brake bands, a high-range clutch and the direct-drive clutch are hydraulically operated as shown in the schematic diagram illustrated in the drawing below.

Turbines and direct-drive clutch are attached to the shaft extending back to the rear sun gear in the planetary. This sun gear meshes with long planetary pinions which, in turn, mesh with short pinions. The short pinions engage the forward sun gear and the ring gear. Output from the planetary unit is taken from the pinion cage. Holding the forward sun gear stationary with the low-range band causes the planet cage to be driven at a reduced speed (1.82 to 1). Releasing the forward sun gear and holding the ring gear with the other band drives the cage in the reverse direction at reduced speed (1.64 to 1). Releasing both the forward sun gear and the ring gear and engaging the high-range clutch locks the planetary train which then rotates as a unit at the same speed as the input shaft.





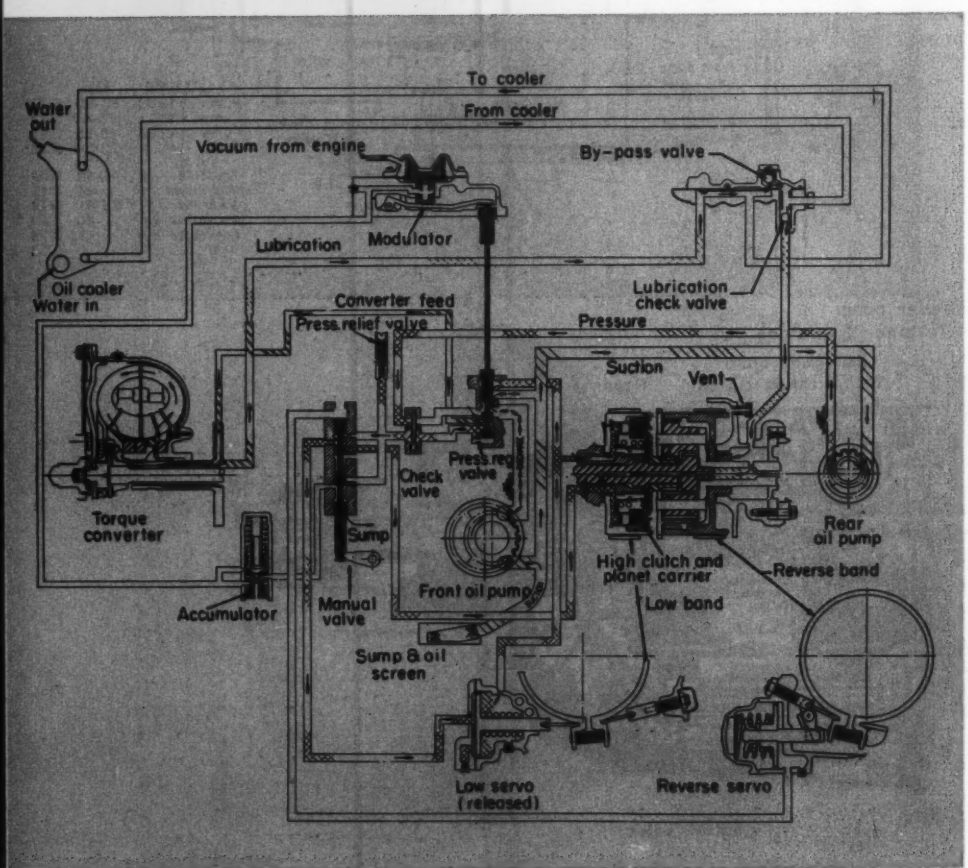
POWERGLIDE



CHEVROLET'S Powerglide automatic transmission, below, is similar in operation to its General Motors predecessor, the Dynaflow, in general design of the torque converter and planetary transmission. Here again, the polyphase converter produces an efficiency curve which reaches higher values at less torque multiplication, obviating the need for a direct-drive clutch. However, the Powerglide version makes use of an overrun coupling, located inside the torus section of the converter, to prevent coasting or freewheeling when the accelerator is released or when on a down grade. Two opposing sets of vanes, one on the turbine and one on the pump, are made up of crescent-shaped vanes or blades so arranged that little oil flow is set up when the pump is rotating faster than the turbine. When the turbine overruns the pump, however, the turbine vanes cause a violent oil flow to be directed against the pump vanes with a consequent torque transfer to the pump —thus braking the engine.

Another important difference in the Chevrolet drive lies in the method of manufacturing the converter elements. While Buick and Packard make turbine, pump and reactor elements from aluminum or aluminum alloy precision castings, the five converter and two overrun coupling members in the Powerglide unit are fabricated from brazed steel stampings.

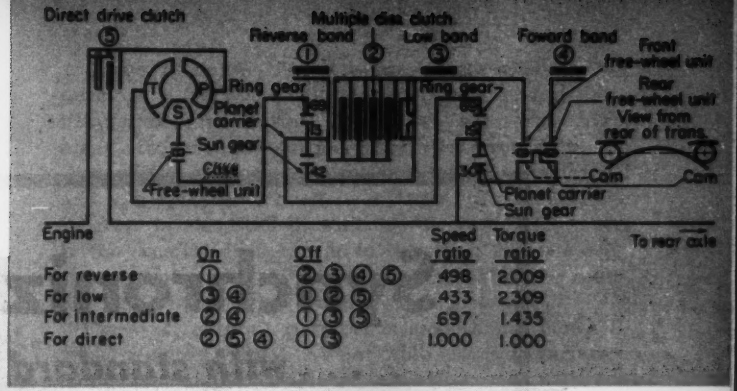
As in the Dynaflow, the Chevrolet transmission provides emergency low and reverse speed reductions of 1.82 to 1, with operation of the disk clutch and low and reverse bands being much the same in both units. Operation of the planetary gearset is also essentially similar. Maximum torque multiplication in the Chevrolet converter is 2.2 to 1.



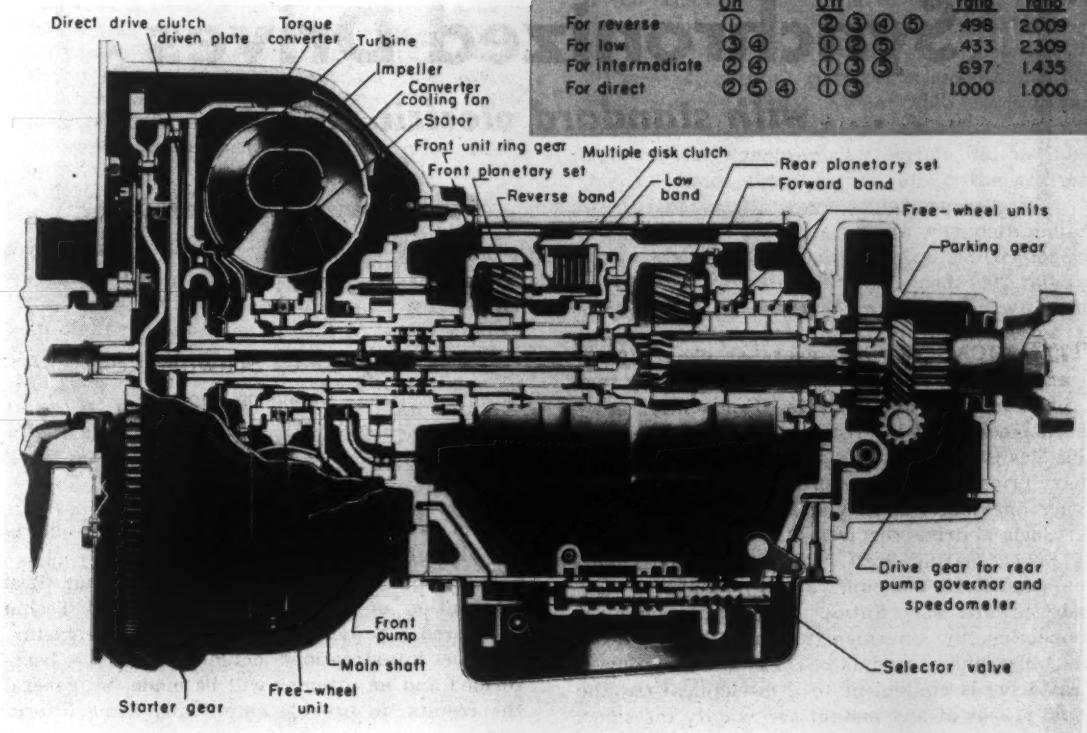
In the control diagram, left, major components of the hydraulic control system are shown in Drive condition. A thermostat in the lubrication oil line operates a bypass valve to admit converter oil to a cooler if temperatures should exceed 240 F. The water-jacket type cooler is interposed in the engine cooling system between radiator outlet and the suction side of the water pump.

As in other automatic transmissions, moving the shift lever to the Park position engages a toothed pawl with a gear on the planet carrier to prevent propeller shaft rotation.

STUDEBAKER



	On	Off	Speed ratio	Torque ratio
For reverse	①	② ③ ④ ⑤	.498	2.009
For low	③ ④	① ② ⑤	.433	2.309
For intermediate	② ④	① ③ ⑤	.697	1.435
For direct	② ⑤ ④	① ③	1.000	1.000



MOST recent of the automatic transmissions to be employed in a passenger car is the unit designed for Studebaker by the Borg-Warner Corp., above. This transmission uses a simple three-element, single-stage torque converter fabricated from stampings, with the uniform-thickness blades held in their housings by a snap fit into detents and bent over lugs on the blades. Two planetary gearsets in series are employed to provide reverse, low, intermediate and direct drive (converter bypassed) speeds. In direct drive a friction clutch is automatically engaged, much as in the Packard drive, to obviate decreased efficiency of the single-phase converter at high speeds.

Operation of the transmission can be followed by referring to the schematic drawing of the gear train, upper right. In low gear, the low and forward bands hold the sun gears of both planetary sets stationary. The converter turbine drives the ring gear of the forward planetary, the output being taken from planetary carrier. This carrier, in turn, drives the ring gear of the rear planetary and, since the rear sun gear is held stationary, power is taken from the rear planetary carrier to drive the propeller shaft. This series operation of the two gearsets gives speed reductions of 1.609 to 1 and 1.435 to 1 which, together with the maximum converter torque multiplication of 2.15 to 1, gives a maximum low gear torque ratio of 4.96 to 1.

In the intermediate range, the front planetary speed reduction is cut out, the low band being released and the multiple-disk clutch applied to lock the planet carrier and sun gear. The rear planetary then functions as before to give a 1.435 to 1 speed reduction. When the direct-drive clutch is engaged, both sun gears freewheel and the gear train is locked out of the drive. All bands and clutches are released for reverse operation, except the reverse band which holds the front planet carrier stationary. The forward sun gear then drives the rear sun gear in the reverse direction through the front unit at a 2.009 to 1 reduction, (4.30 to 1 torque ratio).

The Studebaker transmission has several unique features. A hydraulic interlock prevents engagement of the parking pawl at car speeds above two or three mph. Converter cooling is accomplished by introducing air into the converter bell housing where vanes on the converter shell circulate the air and then expel it from the housing. Accidental shifting to reverse gear is prevented at speeds above 10 mph by a hydraulic lockout which vents the reverse servo line, and a solenoid-operated check valve eliminates traffic light creep by trapping brake pressure to the rear wheels.

Synchronized Drives

. . . with standard electric motors

By E. L. Schwarz-Kast
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University of Chicago
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MECHANICAL shafting, gears or chain drives are conventional means for synchronizing two or more individual shafts. Frequently, however, such mechanical interconnections encounter appreciable design difficulties. A good example is the customary bridge travel drive of large loading bridges, employing one motor arranged at the center of the bridge. Such a drive requires difficult shafting, including bevel gearing with bearings exposed to all weather conditions and difficult to maintain.

Trends toward substitution of these mechanical interconnections by conveniently located electrical devices is very understandable. Such a synchronized electrical drive is equivalent to a mechanical tie, the individual speeds at any instant are exactly the same, irrespective of load differences. In case of different loads there might be an angular displacement between the individual motors, equivalent to the torsion angle of a mechanical shaft.

Two main types of synchronized electrical drives are in use: Selsyn or synchrotie system and simplified self-synchronizing system. The first type, also known as power selsyn, employs a transmitter motor coupled to the main motor. Its speed is the pattern for one or more receiver motors which drive the follower shafts direct. If the follower shafts are driven by individual motors, the receivers are coupled to these motors to synchronize these mechanically independent shafts. Descriptions of this system, its characteristics and the theoretical analysis of operation have been published.[1]

Requires No Additional Transmitters or Receivers

The second method, called simplified selsyn, employs standard wound-rotor induction motors as main motors for each individual drive, using these main motors themselves as selsyn motors without additional transmitters or receivers. This system is also described in literature[2] and has been known for many years. Occasionally an attempt was made to develop an analysis of its operation.[3] Each of these, however, has been extremely difficult to apply. Hence this system is seldom used, even where it could be applied to full advantage.

Recognizing this situation, the Armour Research Foundation of the Illinois Institute of Technology sponsored a project for studying this problem. During the investigations actual tests have been performed and an attempt will be made, by generalizing the results, to provide empirically some information which may serve as a useful guide and facilitate use of the simplified selsyn wherever it is appropriate.

This system uses two or more duplicate standard wound-rotor induction motors. *Fig.* 1 shows the wiring for two such motors. Both stator windings are connected in parallel. Care has to be taken to tie the stator phases together corresponding to the sense of rotation in which the motors operate. The slip rings of both motors are also interconnected, tying the equivalent rotor phases together. Across the rotor connecting leads, as shown in *Fig.* 1, a combined rotor resistor is arranged, which plays an important role in this setup. Without this resistor no synchronized run can be established as will be shown later.

EQUIVALENT ROTOR PHASES: As mentioned, the equivalent rotor phases of both motors have to be tied together. It might be appropriate to describe a practical procedure for identification of these equivalent rotor phases in the field.

1. Connect both stators in parallel to the line, both primaries in the sense of rotation as they are supposed to operate, and leave the rotor leads open. Call them at random r_1, r_2, and r_3 of one motor, and R_1, R_2 and R_3 of the other
2. Connect R_1 with r_1
3. Connect a voltmeter, having a range of twice the rotor voltage, between R_2 and r_2
4. Apply three phases to the stators. Motors will not

[1] References are listed at end of article.

rotate since the rotor circuits are open. Turn one rotor by hand or mechanical means until voltmeter R_2r_2 reads zero

5. With the rotor in this position insert the voltmeter between R_3 and r_3. If the rotor phases are equivalent, this should also read zero. If it does, the interconnection R_1r_1, R_2r_2 and R_3r_3 are correct and final. If not, the voltmeter has to be connected between R_2 and r_3 and the rotor again turned manually until the voltage reads zero. The voltage R_3r_2 should also read zero and a check should prove this. In this case the final rotor connection will be R_1r_1, R_2r_3, R_3r_2. The final rotor positions established by this test are also the synchronized positions, which can be used for the final coupling of the mechanical parts with the motors at zero degree angular displacement.

PERFORMANCE OF DRIVES: Tests were performed with two identical wound-rotor motors, one test series with two ¼-hp four-pole and a second series with two 5-hp eight-pole motors. It had to be anticipated that the results of these two series will be different, since the constant mechanical losses with the smaller motors are of much greater influence than with the larger ones. It can be assumed that the results with the ¼-hp motors will give a general picture of the behavior of fractional horsepower motors and those with the 5-hp motors will be representative of integral motors.

To make the tests more generally usable, torque is expressed as the ratio of developed torque to rated torque and the resistance as the ratio between employed and basic resistance. Basic resistance R_b per rotor phase is expressed:

$$R_b = \frac{E_2}{\sqrt{3}\, I_2} \text{ ohms}$$

where E_2 = open voltage across slip rings and I_2 = rotor current per slip ring at rated full load. For easier identification one of the motors is called the transmitter and the other the receiver.

One rotor blocked: The simplest test for obtaining basic information was to block one motor and measure the torque developed by the free rotor versus angular displacement. This test was made without and again with a resistor 0.23 R_b.

The diagram *Fig. 2* shows, with the transmitter blocked, the torque developed by the receiver versus angular displacement, curve *A*, without resistor, and curve *B*, with a resistor 0.23 R_b. This resistor was found to be the optimum as will be seen later.

Rotor Resistor Is Necessary

These curves give valuable information. In the first place, without resistor and at zero displacement, no torque is developed as shown by curve *A*. Therefore, to make this two-motor drive operate, the insertion of a rotor resistor is necessary. Furthermore curve *A* shows that when the receiver rotor is displaced, appreciable torques are developed which try to bring the rotor back to zero displacement, a fact which can be used for pulling dislocated rotors back into the synchronized position.

Curve *B* shows that with a resistor the receiver develops a substantially higher torque and at zero displacement the torque is not zero, but rather high, about twice rated torque. In other words, if the transmitter is blocked, a resistor 0.23 R_b is inserted and the receiver loaded over twice rated load, the receiver is also stalled, which is desirable. However, if the receiver load torque is less than 200 per cent, the receiver starts rotating in the sense of the rotating field and the synchronization is lost.

Curve *A*, without resistor, is smooth but curve *B*, with resistor, shows remarkable torque variation along the angular displacement. However, with the resistor, the synchronizing torque did not drop below 1.5 times rated torque along the entire range from zero to 360-degree displacement.

Both Motors Running: Tests with both motors running were performed with the transmitter loaded with a constant torque close to full load, 0.97 rated torque with the ¼-hp and 0.85 rated torque with the 5-hp motors. In order to obtain information on the influence of the resistor, the common secondary resis-

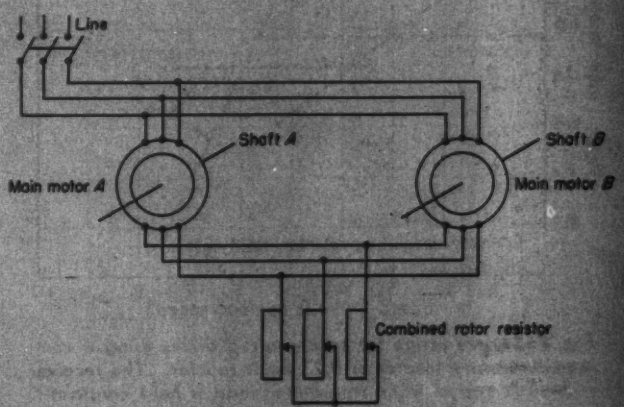


Fig. 1—Diagram for synchronized drive using two duplicate wound-rotor induction motors

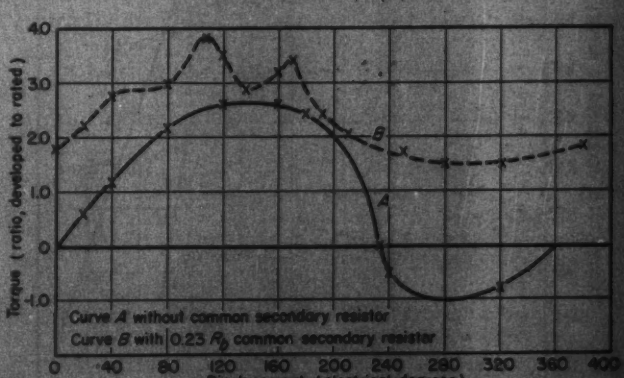


Fig. 2—Receiver torque plotted against angular displacement with the transmitter blocked

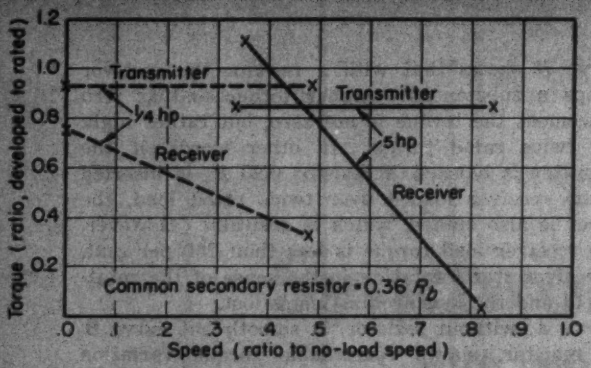


Fig. 3—Synchronous range for transmitter and receiver, using a common secondary resistor of 0.36 basic resistor on ¼ and 5-hp motors. The receiver load is varied and transmitter load held constant

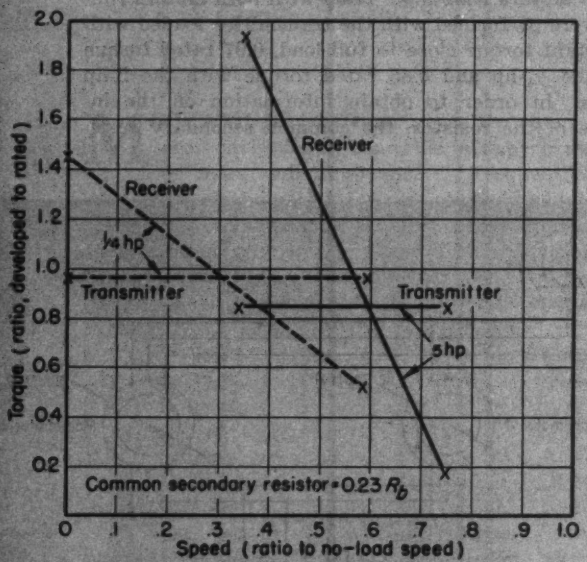


Fig. 4—Torque range for synchronized drives using a common secondary resistor of 0.23 basic resistor. The receiver load is varied while transmitter load is held constant

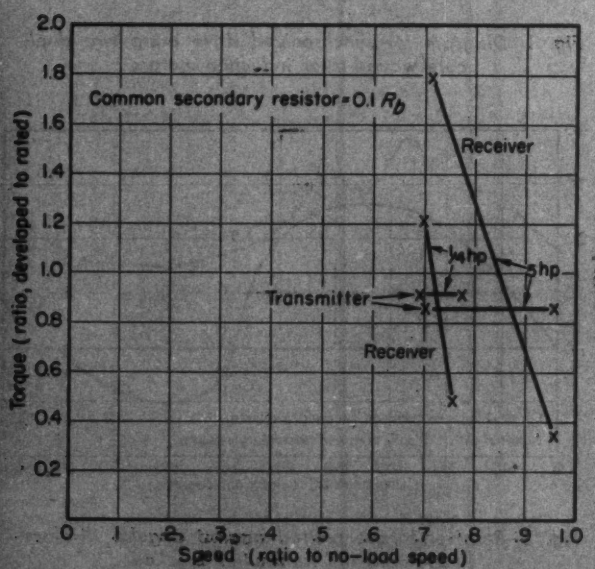


Fig. 5—Torque developed while transmitter and receiver remain in synchronization with a secondary resistor of 0.1 basic resistor receiver load varied, transmitter constant

tor was varied. Resistors equal to 0.36 R_b, 0.23 R_b and 0.1 R_b were inserted in the rotor circuit. The load on the receiver was varied from a minimum to a maximum until synchronous run was lost. If the load was decreased below the minimum the receiver speed increased. If the load was beyond the maximum torque the speed of the receiver dropped out of synchronized run. The test results are shown in *Figs.* 3, 4 and 5.

At this point two extremes should be investigated, resistance infinite (no connection between the three rotor phases) and resistance zero (the three rotor phases short circuited).

With infinite resistance: With both rotors in synchronized position the motors do not start because the rotor voltages of both motors are in opposition, the combined voltage is zero, and no rotor current can flow. The motors remain stalled and the primaries draw only the magnetizing current from the line.

With zero resistance: Both motors behave like two independent squirrel cage motors. No synchronizing torque is developed. Thus, if the loads on both motors differ, their slip will be different and no synchronized run will be established.

Size of Resistors: From the curves in *Figs.* 3 to 5 it becomes apparent that, for the selection of the proper secondary resistance, both the maximum and the minimum receiver loads must be known. In case the minimum receiver torque is zero with the receiver idling, this system cannot be used at all. If the minimum receiver load is low, arround 0.1 rated torque, a higher common resistor, 0.36 R_b is recommended. On the other hand, with this high resistor the maximum receiver torque is rather low, only 1.13 rated load. Also, slip is high and the regulation poor. If the maximum receiver load is high, then a lower resistance is adequate. A value of 0.23 R_b is fairly convenient in both ways. With the transmitter fully loaded, it permits a synchronized run with the receiver loaded between a minimum 0.2 rated load and a maximum 1.95 rated load.

Starting Resistor May Be Employed

The tendency will be to use a secondary resistance no greater than necessary for the maximum and minimum receiver torque in order to keep slip and losses as low as possible. If, with the selected permanent secondary resistor, the inrush current at start is inconveniently high, then an additional secondary resistor can be arranged for starting. It can be assumed that, in general, a synchronized start can be achieved with a resistor of not more than 0.5 R_b. After start, the portion which exceeds the operating value should be shorted.

It might be appropriate to recall that the common secondary resistor and also its common leads carry the sum of both rotor currents and the current rating of resistor and its leads must be twice the full load rotor current of the single motor.

Stopping: Conventionally, the drive is stopped simply by disconnecting the motor primaries from the line. Deceleration is achieved by the load, fric-

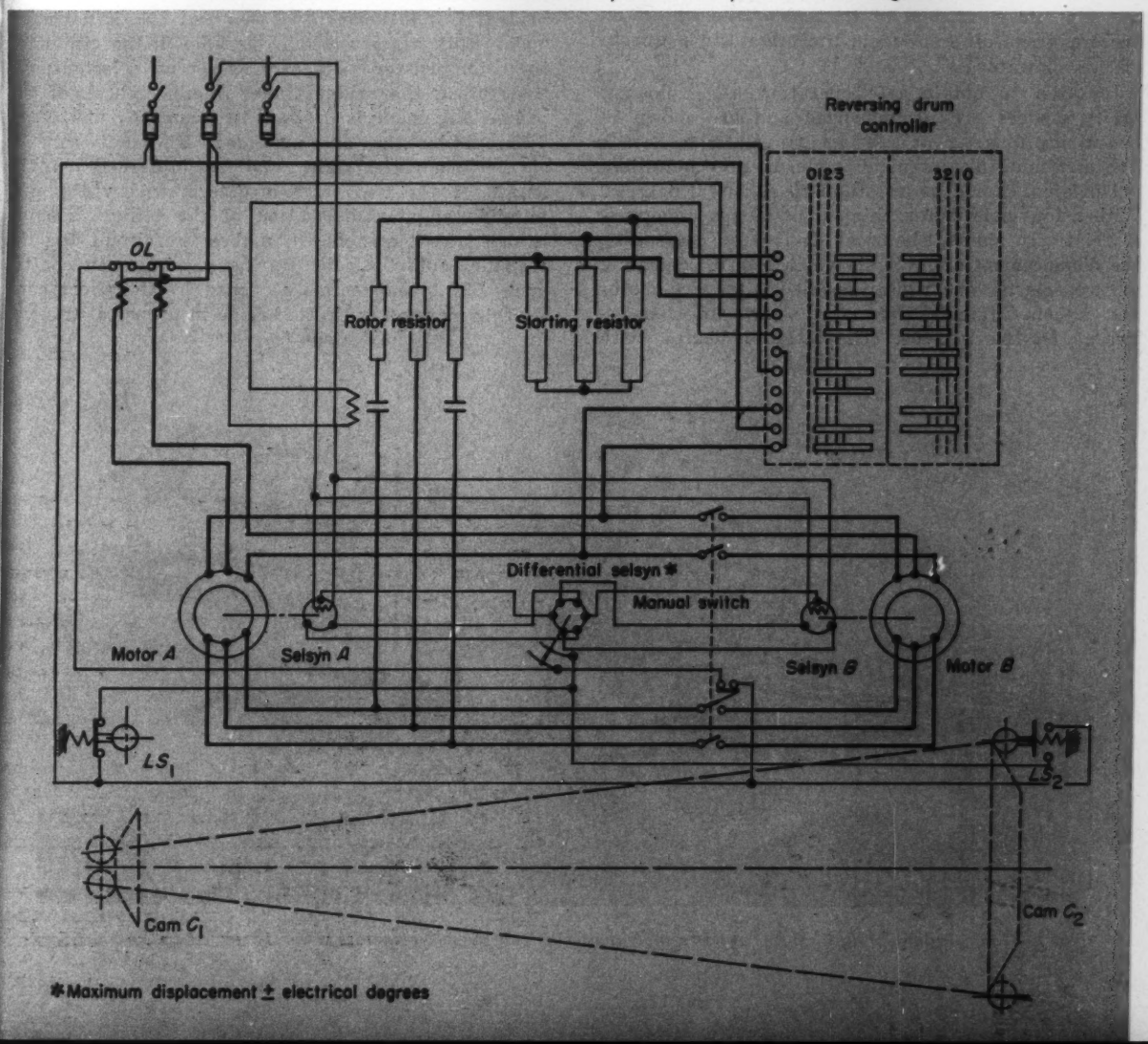
tion, and if necessary, by applying a brake on each motor. This method has a definite disadvantage in that during deceleration the motor primaries are disconnected from the line, and during this period, synchronism is lost. The rotors will synchronize at the next start, but it is very probable that both motors will be in a dislocated position since the number of synchronized positions is equal to half the number of poles, two in the case of four-pole motors.

Moreover, there is a possibility that the dislocation of the drives might accumulate with succeeding stops. Therefore, it is recommended that the stop be accomplished not by disconnecting the primaries, but by opening the common leads to the resistor, leaving the rotors interconnected, and the stators line-connected. The drives come to rest under the influence of load, friction and stopbrakes. By this method, an appreciable synchronizing torque is maintained during the deceleration period. The control device should have a stop point, with the motor primaries line-connected and the resistor circuit open. The operator stops at this point before going to zero and, after the drive comes to rest, he puts the controller at zero. Starting from zero at this point, the drive will not start, but the rotors will synchronize and the actual start will occur at the second step, with the resistor circuits closed. This control arrangement is shown in *Fig* 6.

SHORTCOMINGS OF THE SYSTEM: The user of this method of a synchronized drive should realize the shortcomings in order to avoid failures in operation and to provide the necessary safety devices. In the first place, synchronized run is secured only if the load difference of the two motors does not exceed the permissible limit.

Assuming that the optimum resistor 0.23 R_b is used, the permissible load differences are shown in *Fig*. 4. However, there are very few drives where the actual load situation and the maximum load differences can safely be predicted. After all, even with the loads constant, there are unpredictable accidental changes in the mechanical parts; lack of lubrication, frozen bearings, jammed gears and the like, which might cause a rise of the torque difference and, when

Fig. 6—Wiring diagram for a two-motor bridge travel with automatic alignment control. Phantom sketch shows cam and limit switch positions for permissible misalignment



it exceeds the limit, synchronism is lost.

Another limitation is the stopping. As mentioned, with the conventional method of stopping by disconnecting the primary and applying a brake on each motor, no synchronizing torque is developed during deceleration. A difference in the rate of deceleration of both drives might occur and the displacement might accumulate with consecutive stops. Stopping by opening the rotor resistor improves this situation but there still remains the possibility of a stop due to failure of power supply, where again no synchronizing torque is developed and dislocation may be expected.

SAFETY DEVICES: Due to the shortcomings of this method, the arrangement of safety devices is important in all cases where a high degree of alignment is paramount. These devices must stop the drive at the moment the maximum permissible displacement angle is exceeded, no matter what the cause of the displacement might be. A good example for a two-motor synchronized drive is the bridge travel of a loading bridge or an overhead traveling crane where each leg is driven by an individual motor, without any mechanical connection between the two drives.[4] In this case, even with a perfect synchronized drive, a dislocation of the bridge is possible if, for instance, the slip and coefficient of friction of the wheels on one track are different from the other track, due to oil, ice, or water.

In order to obtain safe operation, the following safety devices are recommended and are shown in the wiring diagram of a bridge travel drive, *Fig.* 6. The maximum displacement angle, to prevent falling out of step, should be less than 80 electrical degrees. With a four-pole motor, this equals 40 space degrees. A relatively simple electrical device for controlling the displacement angle is shown in *Fig.* 6, consisting of two selsyns, one rigidly coupled with each motor and a conventional differential selsyn inserted between the two selsyns. The latter indicates, with sufficient accuracy, the angular displacement of the two rotors and also develops ample torque for control purposes. For example, the differential selsyn can actuate a sliding contact which interrupts the coil of the resistor contactor when the displacement angle exceeds the preset limit in either direction, thus stopping the drive in synchronism.

In general, it will be sufficient to arrange some kind of alignment indicators somewhere along the bridge travel, thus permitting the operator a visual check of dislocation. An automatic alignment control can also easily be arranged by limit switches as shown in *Fig.* 6.

Two limit switches, LS_1 and LS_2, are located on the bridge; LS_1 is normally closed and LS_2 is normally open. Both are inserted in parallel in the circuit of the rotor-resistor contactor. At several selected positions along the bridge travel where a check of the bridge alignment is desired, two cams C_1 and C_2 of different lengths are arranged. The left cam is shorter than the right, and the difference in the length of the two cams is determined by the permissible angular dislocation of the bridge. During bridge travel, switch LS_1 makes the circuit for the resistor contactor coil. During passing each control spot, LS_1 is momentarily opened by the short cam. At this moment, with the bridge in proper alignment,

(*Continued on Page* 234)

High-Speed Drives Reprint Available

If you would like to have a personal copy of the special editorial section on high-speed drives, a company-letterhead request will bring one with the compliments of the Editors of MACHINE DESIGN. If quantities are desired, it will be a pleasure to quote prices.

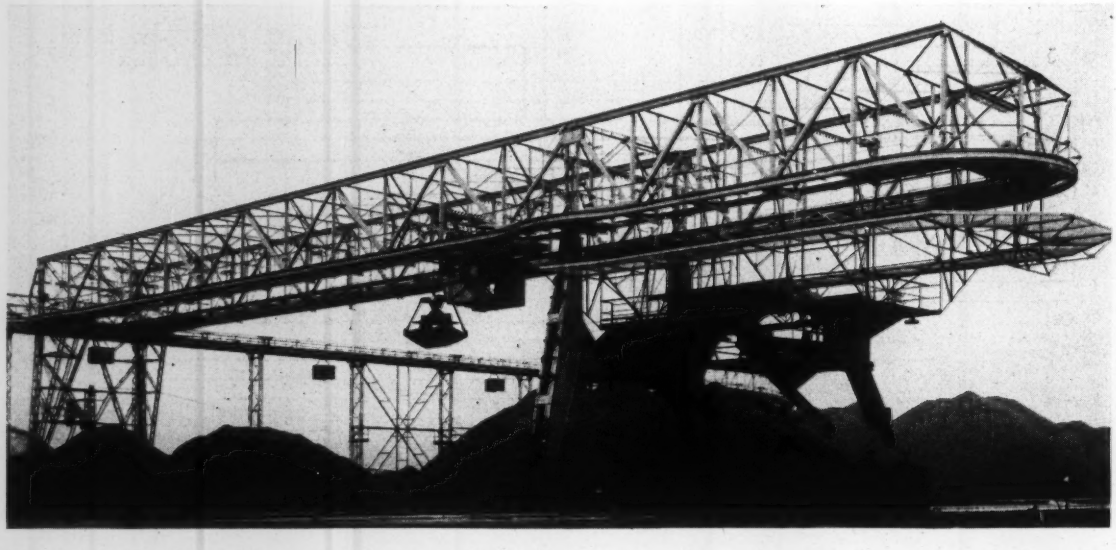


Fig. 7—Coal loading bridge. Both legs are individually driven by selsyn motors without mechanical connections

High Speeds... IN DESIGN

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MACHINE DESIGN
APRIL, 1950

A SPECIAL MACHINE DRIVES AND CONTROLS PRESENTATION

High Spe

THIS EDITORIAL PRESENTATION is devoted specifically to the problem of attaining the desired speed of rotation for a particular machine drive. Basic methods of high-speed drive are discussed and their primary characteristics are outlined and developed to provide an overall present-day picture of practical attainable speeds. Advantages, limitations and design requirements are presented in the general order of increasing speed to permit an integrated analysis of this interesting and important basic consideration in the design of all types of machines

How high rotary speeds are developed and how to select and design for the most satisfactory drive

...ds...

PERHAPS the questions, "What speed?" and "Is it practically attainable?" are among the most common and fundamental which confront the machine designer. In many cases too little consideration is given to these problems of drive, stemming undoubtedly from the apparent simplicity. Most satisfactory results and, more important, most satisfactory service life makes adequate analysis imperative. This is especially true where power transmitted is to be high or speeds necessary or desirable range upward from the normal encountered in general applications.

Machine drives may take numerous forms but in overall analysis they most generally fall into one of several categories. These, not including internal-combustion engines, turbines, etc., might be roughly divided into direct motor drive, gear, chain, or belt. Direct motor drives might be subdivided into electric motor, pneumatic motor, hydraulic motor, or air turbine. Chain, of course, would subdivide into roller and silent; belt into leather, V-belt and flat fabric. The problem then is, where does each fit into the integrated picture of drive with reference to speeds practically attainable? The purpose of this special presentation is to provide this general design study with the hope of facilitating the selection of the most suitable drive and proper consideration of its application to a particular machine.

HIGH SPEEDS: Today, it is seldom necessary to sacrifice any fundamental functional need owing to inability to obtain the necessary speed of rotation. Actually, no absolute top speed has been reached to date. The top limit on revolutions per minute is set mainly by the practical factors of materials and mechanical design to withstand the loads involved and the cost. An overall picture of speeds is delineated in the chart of *Fig.* 1.

Naturally, speed of rotation is not the sole selection factor; horsepower available or possible to transmit

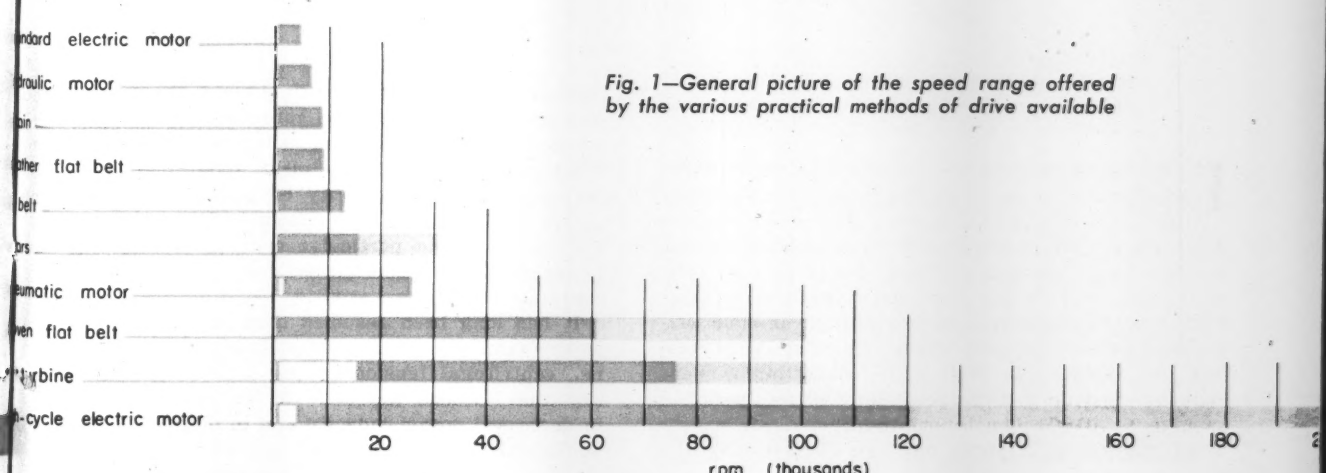


Fig. 1—General picture of the speed range offered by the various practical methods of drive available

at any particular speed varies widely, among the several means which can be called into play, over a large portion of the total speed range. Each method of drive must be considered separately on the basis of its own particular characteristics and, consequently, power is not brought into the chart of *Fig.* 1 owing to the many variables involved. In all probability, the high-frequency or high-cycle electric motor provides the maximum in horsepower per pound of drive wherever direct drive is possible or advantageous, *Fig.* 2.

DESIGN: In the overall design of a drive for speeds over and above conventional speeds, a number of factors must be considered. These design factors generally might be enumerated as:

1. Speed required
2. Horsepower necessary
3. Space available
4. Drive suitability
5. Cost.

Each of these factors must be considered fully in selecting the final drive medium. Failure to include one factor may result in an unsatisfactory drive for any of numerous reasons. From the data outlined in the subsequent portions of this section, a reasonable overall analysis of all these design factors should be possible. Arrival at the most practical drive to suit any speed-horsepower requirements as well as prevailing operational conditions, it is sincerely hoped, will be greatly simplified.

The final factor of cost, of course, is not unimportant. In many cases, the mere fact that optimum speed-horsepower requirements can be met in a practical manner reduces the significance of drive cost in overall machine cost. However, in the main cost *is* important and generally must be justified.

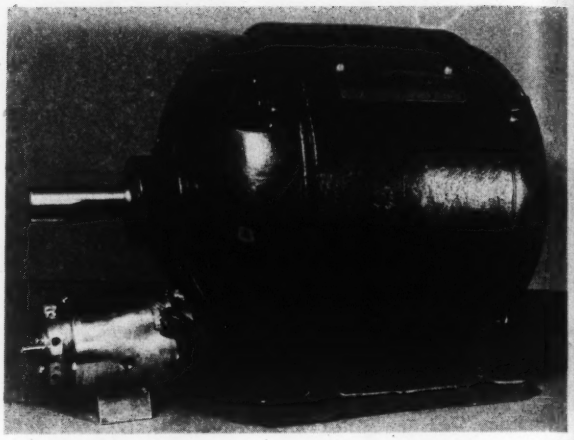


Fig. 2—Comparison in size of a standard 3-hp, 1800-rpm electric motor weighing 105 pounds and a 3-hp, 2000-cycle, 120,000 rpm motor which weighs only 7 pounds

Owing to the many basic design differences in the field of high-speed drives, a direct cost comparison is impossible and perhaps even undesirable. Wide selection is practical only at the lower end of the speed range and beyond 20,000 to 30,000 rpm the choice narrows drastically, *Fig.* 1. Over 70,000 to 80,000 rpm, only the high-cycle motor remains as a practical drive. Thus, even though a cost-per-horsepower or cost-per-rpm comparision might be made, its usefulness and significance in the light of practical application would be nil. Minimum cost in the final analysis can be achieved most readily by skillful design within the range of the particular type of drive dictated by the major machine requirements.

Chains

IN seeking a method for transmitting up to 5000 horsepower, or an exact speed ratio between two parallel shafts, a high-speed roller or silent chain drive should not be overlooked as a possible answer to this design problem. Chain drives provide efficiencies of over 99 per cent and have been successfully used for rotational speeds as high as 8000 revolutions per minute with chain velocities above 8000 feet per minute. In *Fig.* 3 a $\frac{3}{8}$-inch pitch silent chain is used on a high-speed hand plane operating as high as 10,000 rpm intermittently.

Advantages of stepping up chain velocities to the highest value consistent with desired chain life are obvious—a spindle can be rotated faster and more power can be transmitted using a smaller chain. What sets the limit on lineal speed? Usually, a decrease in power capacity beyond a certain point accompanied by excessive chain and sprocket wear. For instance, the particular chain shown in *Fig.* 3 is generally not recommended for over 6000 rpm for continuous service.

It has long been assumed that chains are subject to the same forces that affect belt and cable drives, i.e., centrifugal tension causes a stretch or elongation that loosens the belt on the pulleys resulting in decreased power capacity and to offset this effect, additional tension must be applied to belt drives

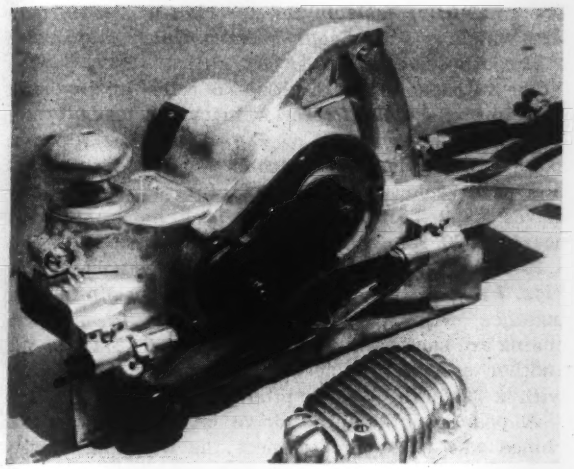


Fig. 3—Above—Hand plane employing 3/16-inch pitch silent chain for intermittent spindle speeds to 10,000 rpm. Power involved is about 1 hp and a light grease is used for lubrication

(there is a speed where this additional tension is sufficient to decrease belt life and increase bearing wear to the point where the drive becomes impractical). With chain drives, however, centrifugal tension does not limit power transmission. Traction is obtained through tooth engagement. In fact, maximum chain efficiency is obtained when centrifugal tension equals the working load.

Properly designed chain drives stand up well under high loads for long periods. A high-speed newspaper press equipped with four 100-hp silent-chain drives with ⅝-inch pitch, 6-inch wide chain operating over 44-tooth sprockets on 28¾-inch centers at a maximum chain speed of 4700 fpm, has given satisfactory operation for over ten years. As another example, a hammer mill powered by a 75-hp, 1750-rpm motor and equipped with 4-inch wide, ½-inch pitch silent chain operates satisfactorily at a chain speed of 5400 fpm. In another instance, the chain on a front-wheel driven truck is required to run at 8000 fpm in high gear and still transmit full rated engine power in low gear.

Roller Chain Carries High Loads

In general, roller chains will transmit higher loads per inch of width than will silent chain. TABLE 1 shows horsepower ratings for standard single-strand roller chain No. 35 having a ⅜-inch pitch. These values are based on a life expectancy of approximately 15,000 hours under optimum drive conditions and a service factor of one. Ratings are for two sprockets; idler and additional sprocket may decrease the rating. TABLE 2 shows part of the tabulation of horsepower ratings for ⅜-inch pitch silent chain. Ratings are per inch of width, with a maximum width of 4 inches in this chain size.

SILENT-CHAIN DRIVES: The ideal high-speed silent-chain drive is considered to conform to the following general characteristics:

1. Small sprocket of 35 teeth, minimum
2. Large sprocket of 160 teeth, maximum
3. Chain velocity about 4700 fpm
4. Working load equal to the centrifugal tension ($WV^2/116{,}000$ where W = weight of chain, per foot, V = speed, fpm)
5. Sprockets balanced, heat-treated
6. Chain case oiltight with a splash oiler.

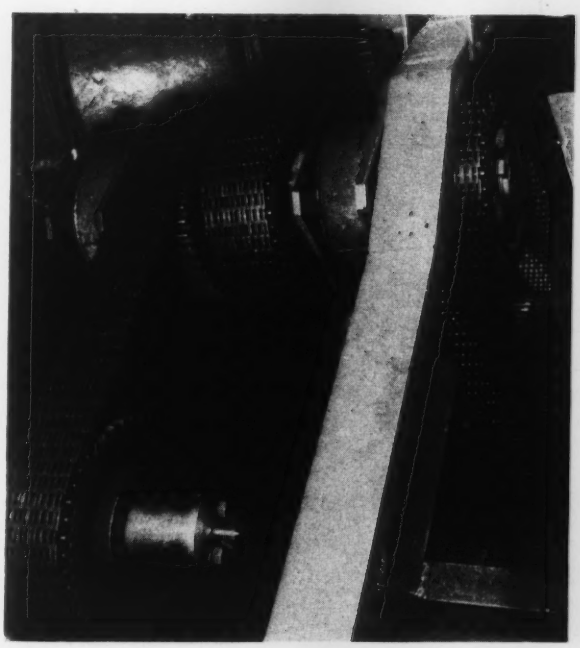
Capacity of a drive with all of the foregoing characteristics proves to be about three times that of a similar drive of the same pitch and width using a 21-tooth pinion. The answer is probably due to three factors:

1. Higher speed with correspondingly less pull on the chain
2. Less chordal action because of greater number of teeth
3. Stabilizing effect of centrifugal tension in the wrapped portion of the chain.

Sprocket wear will not reach troublesome proportions if heat-treated or flame-hardened alloy-steel sprockets are used. Using semisteel, sprocket life will be less than chain life, usually, but with speeds under 4500 fpm such sprockets are practical. The smallest of unheat-treated sprockets should have at least 75 teeth. Use of the largest practicable number of teeth will result in less wear on both chain and sprockets.

Recent practice favors the use of large-diameter sprockets with small-pitch chain. *Fig.* 4 shows such an installation with ½-inch pitch, 2½-inch wide silent chain. On the sprockets shown, which are from 6

Fig. 4—Below—Silent chain drive showing trend toward large-diameter sprockets with small-pitch chain. This drive uses ½-inch pitch, 2½-inch wide chain; 6 to 8-inch diameter sprockets; transmits 75 hp at 3600 rpm



to 8 inches in diameter, this drive can transmit 75 hp at 3600 rpm. The equivalent width chain is ⅜-inch pitch could operate at a speed of 5000 rpm and handle 50 horsepower.

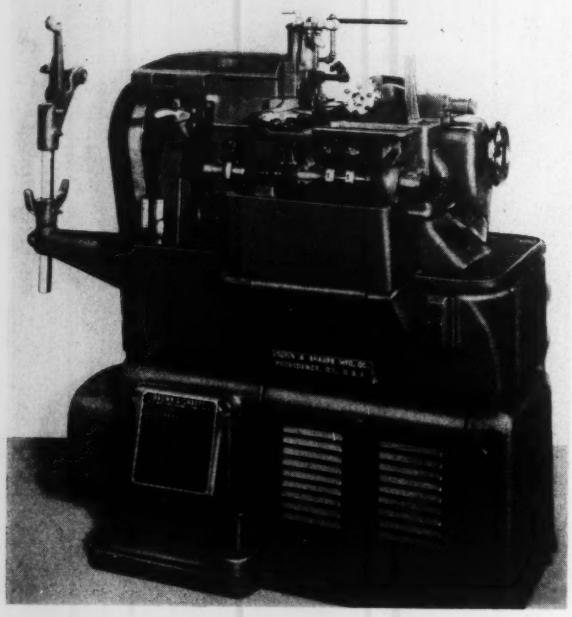
ROLLER-CHAIN DRIVES: Rotative speed and number of teeth on the smaller sprocket are controlling factors in the selection of roller-chain drives. Lineal speed is not a primary limiting factor in relation to proper roller chain selection except as it contributes to centrifugal stresses and pressures. Roller chain is so light in relation to power transmission capacity and strength that centrifugal stresses are of no importance in relation to chain strength until linear speed approaches 30,000 fpm.

High-Speed Chain Fatigue Resistant

High-speed chain is usually less subject to fatigue failure of the links than medium or low-speed chain because the former drives generally have a large number of teeth in the sprockets. Chordal action, or excessive flexing of the chain, is thus reduced to a nearly negligible factor. Also, joint articulation and "worry" are reduced because of the large number of sprocket teeth and the correspondingly low chordal "rise" and "fall" of the chain.

Roller chain manufacturers recommend no less than 15 to 24 teeth on sprockets for high-speed service. Chain action over sprockets having less than this minimum number of teeth cannot be quiet and smooth. It is considered economical to use heat-treated sprockets when the number of teeth is less than 24 and sprocket speed is over 600 rpm.

A high-power drive is shown in *Fig.* 5, as applied to a drilling rig. The 1¾-inch quadruple-strand roller chain seen at the extreme right in the photograph transmits 650 hp from a 900-rpm diesel to a countershaft driving a circulating mud pump at two speeds. Maximum chain speeds are about 3600 fpm.



The final drive is from the sprocket at the extreme left, which is a 1½-inch, 6-strand roller chain using a 31-tooth drive sprocket.

The introduction of high-speed roller chain drives to a line of automatic screw machines made by the Brown & Sharpe Mfg. Co., *Fig.* 6, was the result of extensive testing and research work and collaboration with various chain manufacturers. Satisfactory drives have been secured with chains running at nearly 5500 fpm. In the machine shown in *Fig.* 6, a 29-tooth, ⅜-inch pitch sprocket drives roller chain, *Figs.* 7 and 8, at approximately 5480 fpm. In another instance, with a 41-tooth, ⅜-inch pitch sprocket, chains are successfully operated at 5424 fpm. In still another case, a top speed of 4915 fpm is obtained with a 39-tooth, ½-inch pitch sprocket.

A positive high-speed drive on these screw machines was necessary to meet the widening variety of new materials being produced on which higher machining speeds could be used advantageously. It was believed that a chain drive, properly applied, would permit greater utilization of the machine capacity, provide the necessary additional power desired and produce a drive which would operate with greater reliability and economy than had been previously possible. The belt life on this size machine

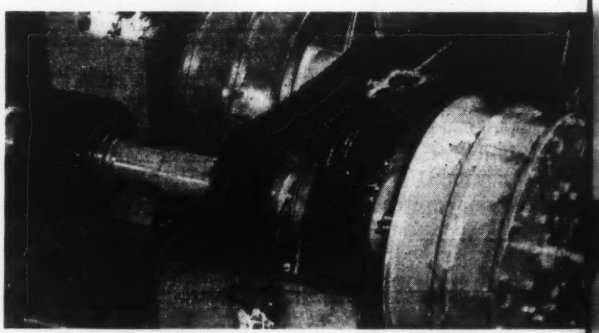
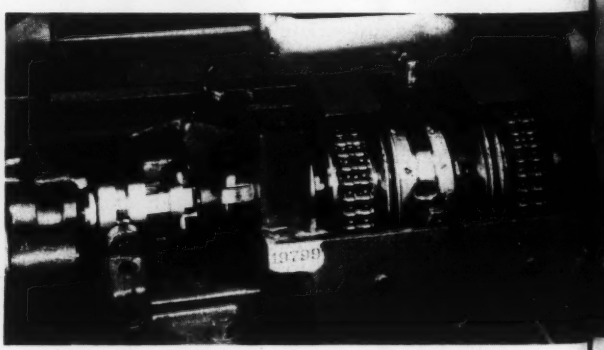


Fig. 5—Above—Quadruple-width, 1¾-inch pitch roller chain on drilling rig. Chain at extreme right transmits power from 650-hp, 900-rpm diesel to circulating mud pump

Fig. 6—Left—Model OOG automatic screw machine on which roller chain drive of Figs. 7 and 8 is used

Fig. 7—Below—Automatic screw machine spindle sprockets on double ⅜-inch pitch chain drive. Friction clutch is visible between the 6050-rpm sprocket on the left and the 3780-rpm sprocket on the right



was normally about 400 to 500 hours, requiring replacement approximately every eight or ten weeks.

It was realized that there were many design problems attendant to such a drive that required consideration in order to obtain satisfactory operating conditions. A chain drive to the spindle had to satisfy the following conditions:

1. Load intermittent, variable, and subject to reversal shock
2. Drive shafts with fixed center distance
3. Idler sprocket to provide compensation for wear and maintain proper tension
4. Drive from electric motor via gearing and chain
5. Two chains to run in same *or* opposite directions
6. Drive vertical with driving sprocket below driven
7. Speeds at spindle to vary from 25 to 3000 rpm.

To be considered worthwhile, the chain drive should:

1. Provide chain life substantially greater than belt life
2. Provide adequate and continuous lubrication
3. Keep heat at a minimum.

Preliminary tests progressively indicated that a chain drive had considerable promise; it had shown itself capable of delivering power in excess of that available with a belt drive, and with better lubrication and the elimination of one idler, the desired speed of 3000 rpm could be maintained. It was found that ⅜-inch pitch triple chain with its reduced impact load promised ideal results.

At high-speeds the dynamic loads on ½-inch and ⅜-inch pitch chains are most important factors. The loads on a chain in order of importance may be considered as (1) Impact load, (2) centrifugal force, and (3) idler tension. The smaller pitch chain is lighter and therefore has a lower impact velocity. Since centrifugal force is a function of chain weight and speed, using a ⅜-inch pitch chain reduced the

Fig. 8—Drive end of chain shown in Fig. 7. High-speed drive sprocket is at the center with the low-speed sprocket just visible at the extreme right of the gear box

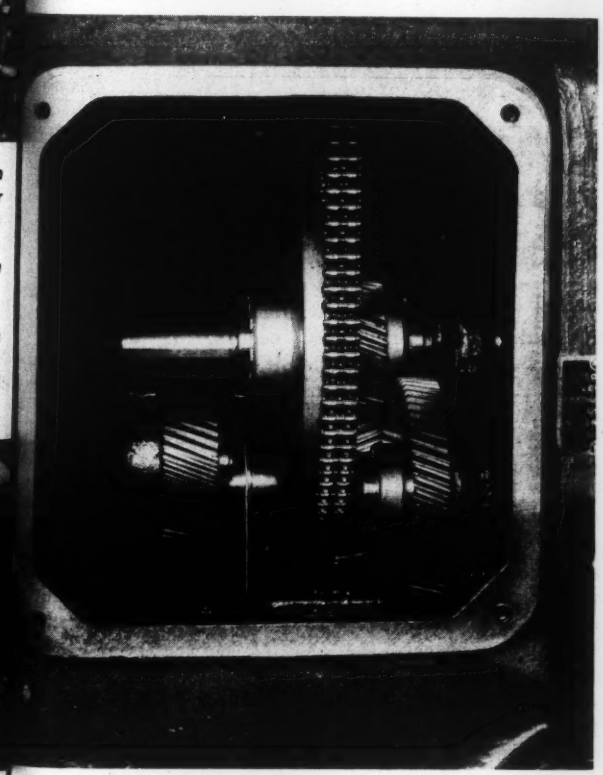
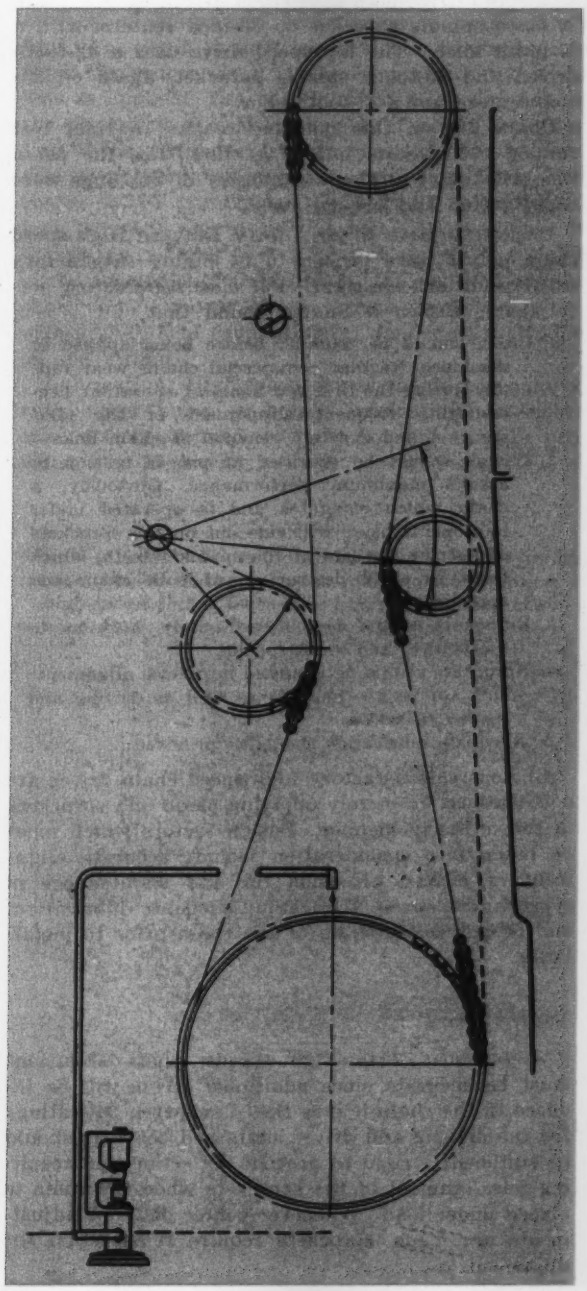


Fig. 9—First chain arrangement used in chain drive for automatic screw machines



centrifugal force 22 per cent below that of a ½-inch pitch chain.

A machine operated on a three-second reversing cycle offers good evidence. The chain used was double ⅜-inch pitch, *Figs.* 7 and 8, as on the previous machine. *Fig.* 7 shows the spindle sprockets and *Fig.* 8 the drive sprockets used to drive the spindle at a maximum speed of 6050 rpm for the high-speed side and 3780 rpm for the low-speed section. By means of a friction clutch between the two spindle sprockets it is possible to select either of the two speed ranges in forward or reverse. The most severe service on the chains exists when they shift from 6050 rpm forward to 1204 reverse and back to forward—all in less than three seconds. The high-speed drive uses a 78-tooth driven sprocket and 29-tooth spindle sprocket on 35-inch centers, with a 37-tooth idler. The low-speed drive uses a 42-tooth driven and 35-tooth spindle sprocket, again on 35-inch centers, and a 23-tooth idler.

Chain life on this machine reached a final test run of 4360 hours, total. At that time the chain was still intact although a number of bushings were badly pitted and broken away.

DESIGNING HIGH-SPEED CHAIN DRIVES: High-speed chain drives have proven to be highly satisfactory when conditions are right. For most satisfactory performance, Brown & Sharpe found that

1. Chain should be "run-in" before being applied to machines. Regular commercial chains wear rapidly during the first few hours of operation, necessitating frequent adjustment of the idler sprocket and eventual removal of chain links
2. Chains should be operated at proper tension to assure maximum performance. Obviously, a chain which elongates and is operated under improper tension will ride out on the sprockets and strike the tips of the sprocket teeth, which leads to early destruction of both chain and sprockets
3. Sprockets should be cut accurately, both as to tooth form and tooth spacing
4. Sprockets should be mounted in precise alignment; this applies to the idler as well as driving and driven sprockets
5. Adequate lubrication should be provided.

Obviously, satisfactory high-speed chain drives are not obtained by merely applying chain and sprockets in the ordinary manner. Design factors which must be taken into consideration include accurate alignment of shafts, provision for and maintenance of correct tension of the chain, adequate lubrication, and proper conditioning of the chain prior to installation.

Shaft Alignment Critical

Alignment: With high speeds, shaft alignment must be accurate since additional stress will be induced in the chain if it is flexed sidewise. Mountings for the driving and driven units and bearings should be sufficiently rigid to prevent deflection and resulting misalignment of the sprockets when the chain is placed under load. Whenever center distance adjustments are made, sprockets require reinspection for alignment.

Table 1—Horsepower Ratings for Roller Chains*

(per single strand of ⅝-inch, #35 chain)

Teeth Small Sprocket	Small Sprocket Speed (rpm)						
	2000	2200	2400	2800	3200	3600	4000
11	1.40	1.41					
12	1.66	1.69	1.72				
13	1.91	1.96	2.02	2.07			
14	2.15	2.23	2.29	2.34			
15	2.37	2.46	2.54	2.65	2.70		
16	2.58	2.70	2.79	2.92	2.98		
17	2.79	2.92	3.03	3.20	3.27	3.30	
18	2.98	3.12	3.24	3.43	3.52	3.57	
19	3.19	3.34	3.47	3.66	3.80	3.83	3.84
20	3.37	3.51	3.65	3.87	4.01	4.07	4.06
21	3.53	3.71	3.86	4.08	4.22	4.28	4.28
22	3.73	3.89	4.04	4.28	4.43	4.47	4.50
23	3.89	4.06	4.22	4.45	4.60	4.68	4.71
24	4.04	4.23	4.38	4.65	4.81	4.86	4.87
25	4.19	4.39	4.54	4.82	4.98	5.06	5.04
30	4.90	5.09	5.28	5.55	5.70	5.73	5.65
35	5.47	5.69	5.87	6.10	6.19	6.14	6.00
40	5.96	6.11	6.32	6.51	6.58	6.60	
45	6.37	6.56	6.68	6.78	6.70		
50	6.71	6.84	6.94	6.89			
55	6.96	7.06	7.05				
60	7.14						

* Courtesy Diamond Chain Co. Inc.

Table 2—Horsepower Ratings For Silent Chain*

(per inch of width of ⅜-inch pitch chain)

Teeth Small Sprocket	Small Sprocket Speed (rpm)								
	2250	2500	2750	3000	3250	3450	3750	4000	4250
17	3.6								
19	4.4	4.7	5.0	5.2					
21	5.1	5.5	5.8	6.2	6.5	6.7	7.1		
23	5.7	6.2	6.6	6.9	7.2	7.5	7.8	8.1	8.3
25	6.7	7.2	7.6	8.0	8.4	8.6	8.7	9.2	9.5
29	7.6	8.2	8.7	9.2	9.6	9.8	10.0	10.7	11.0
31	8.2	8.9	9.5	10.2	10.6	10.8	11.2	11.9	12.2
35	9.5	10.2	10.8	11.6	12.2	12.4	13.1	13.8	14.2
39	10.2	10.9	11.5	12.2	12.7	13.0	13.7	14.1	14.4
45	12.0	12.8	13.9	14.7	15.3	15.7			
55	14.2	15.1	15.8						
65	16.0								

* Courtesy Morse Chain Co.

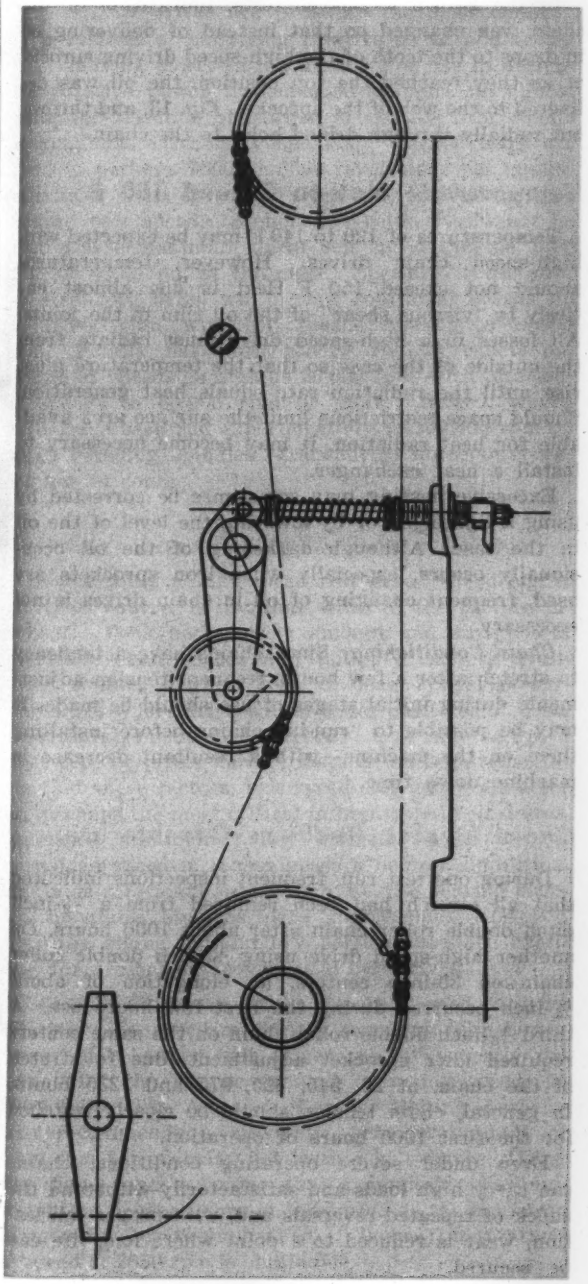
Tension adjustment: Provision must be made for initial adjustment of chain tension and for "take-up" during early life of a drive for the stretch or pitch elongation that inevitably occurs. If adjustable centers are not practicable, it may be possible to obtain the necessary chain adjustment by removing links. Silent-chain idlers, if necessary, should always contact the inside of the chain. For general service, shoes running on the back of the chain generally are better than small idlers, which cause joint wear due to the sharp bend in the chain. Shoes cause no appreciable wear if made of cast iron or wood, and are relatively quiet. To minimize reverse bending and consequent decreased chain life, the minimum shoe radius recommended by chain manufacturers should be observed.

Idler sprockets are not required for normal roller chain drives, but adjustable idlers may be used to provide a practical alternative means for taking up chain wear if adjustable shaft centers are not feasible,

or to guide the chain clear of interferences. It is preferable that idlers mesh with the outside of the slack span and be near the smaller sprocket of a drive. At least three teeth of idler sprockets should mesh with the chain and speed of rotation should never exceed that allowable for other loaded sprockets.

The first chain drive layout used by Brown & Sharpe required two idlers to clear the machine frame, *Fig.* 9. It was found that by redesign and by increasing chain tension, one idler could be eliminated and the chain would still clear the machine frame, *Fig.* 10. *Fig.* 11 shows the final arrangement of the idler tension adjustment of both high and low-speed chains.

Fig. 10—Improved model of test machine incorporated greater chain clearance which, together with increased chain tension, eliminated one idler

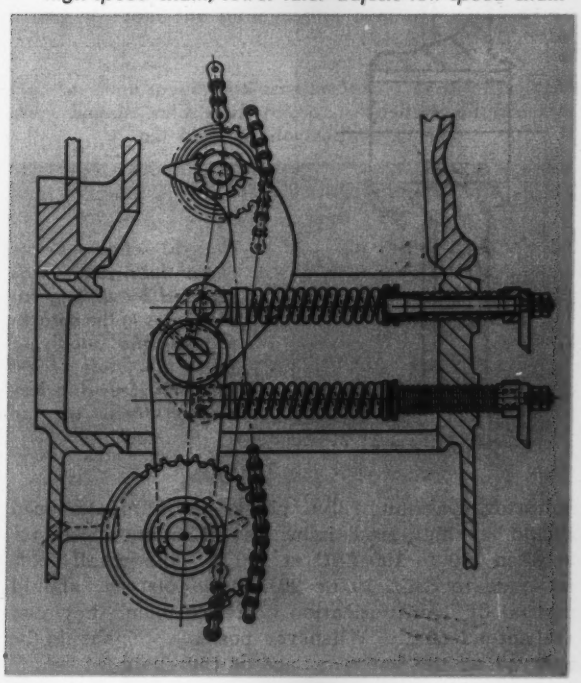


Where zero-backlash is important, a preload about equal to the working load should be imposed on the chain. Then, as the chain is loaded, the nonpulling or slack side of the chain will be released sufficiently to run smoothly and still be free from slap.

On a drive with fixed centers or one on vertical centers, chains should be adjusted to be relatively taut. The chain is sufficiently elastic to stretch and form the proper slack in the idler strand when the load is applied. The best initial tension depends on the character of service, length of chain and position of the particular drive in question. As a general rule to follow, one manufacturer recommends that chains be run at a tension which would be considered just too slack for a leather belt. Another rule of thumb: On a drive having normal centers, a suitable initial tension is determined by the point where the chain can be moved about ½-inch in each direction with a 25-lb load (approximate) applied at the center of one strand.

Lubrication: High-speed drives should be equipped with either pump or splash lubrication. In either case, a fully-enclosed, oil-tight housing must be provided, *Fig.* 12, having suitable bearing seals, oil level

Fig. 11—Correct idler tension adjustment given by spring-loaded device shown. Upper idler controls tension in high-speed chain, lower idler adjusts low-speed chain



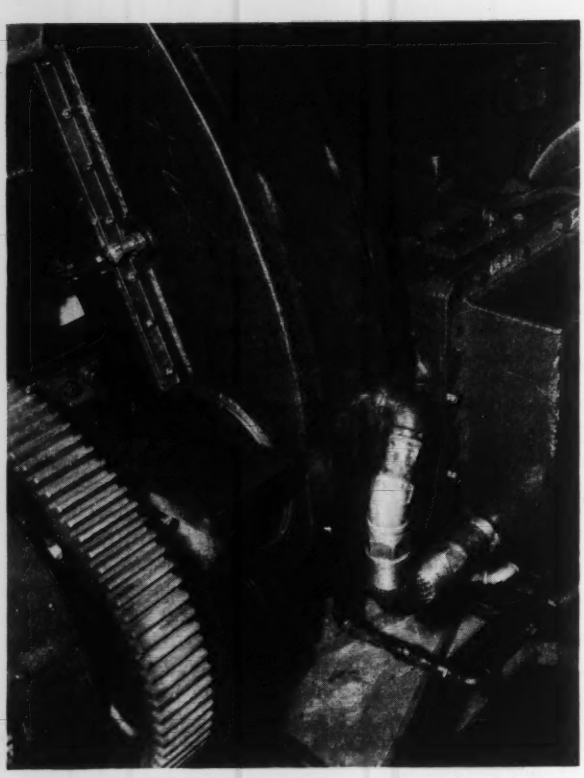


Fig. 12—Above—Essential component of any high-speed chain drive is an oiltight case, such as the one shown, having the necessary seals, oil level indicators, filling and drain plugs

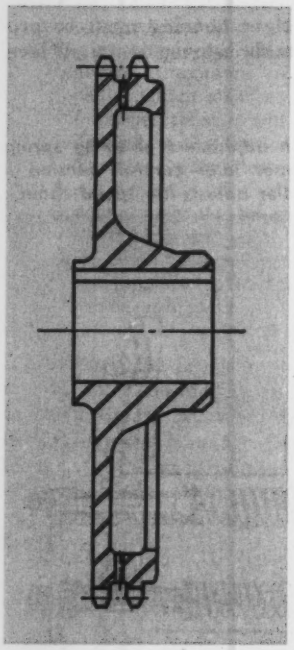


Fig. 13 — Left — Better chain lubrication was realized by drilling radial holes in the sprocket rim and supplying oil to the inside of the rim. Previous drip lubrication threw off too much oil

indicator, drain and filling plugs, etc. Lubricant used should be a high-grade industrial oil having a viscosity between 100 to 150 SSU at 100 F. This usually corresponds to SAE 10 or 20. In *no instance* should manual or drip lubrication be relied upon, they just are not adequate. Whenever possible, oil should be sprayed on the inner surface of the chain.

If pressure lubrication is used, it may be necessary in some cases to bypass some of the oil to prevent an excessive amount from being thrown off the chain. Excessive lubrication is indicated by leakage of oil around the shaft seals.

When the splash system of lubrication is used, the oil level in the chain case should be adjusted so that it is barely touching the chain. It will be found that sufficient oil will be picked up by chain even when oil level is below the lower strands of chain. Actually, "windage" of the chain and sprocket creates ripples of oil in the sump and the crests of these ripples are scanned by the lower strands of chain. Here again, too much oil will cause leakage and sometimes excessive heating.

In the Brown & Sharpe tests, lubrication of the chain was changed so that instead of delivering oil in drops to the teeth of the high-speed driving sprocket as they reached the top position, the oil was delivered to the web of the sprocket, *Fig.* 13, and thrown out radially through drilled holes to the chain.

Temperature Not to Exceed 150 F

Temperatures of 120 to 140 F may be expected with high-speed chain drives. However, temperatures should not exceed 150 F. Heat is due almost entirely to "viscous shear" of the oil film in the joints. All losses in a high-speed drive must radiate from the outside of the case so that the temperature must rise until the radiation rate equals heat generation. Should space restrictions limit the surface area available for heat radiation, it may become necessary to install a heat exchanger.

Excessive heating may sometimes be corrected by using a lighter oil or by lowering the level of the oil in the case. Although darkening of the oil occasionally occurs, especially when iron sprockets are used, frequent changing of oil in chain drives is not necessary.

Chain Conditioning: Since chains have a tendency to stretch after a few hours, frequent tension adjustments during initial stages of use should be made. It may be possible to "run-in" chains before installing them on the machine—with a resultant decrease in machine down time.

Chain Stretch Requires Consideration

During one test run, frequent inspections indicated that all stretch had been removed from a ½-inch pitch double roller chain after about 1000 hours. On another high-speed drive using ⅜-inch double roller chain on 35-inch centers, an elongation of about ¼-inch occurred during the first few hours use. A third ⅜-inch double roller chain on the same centers required idler sprocket adjustment, due to stretch of the chain, at 20, 545, 895, 975 and 1235 hours. In general, chain tension should be closely watched for the first 1000 hours of operation.

Even under severe operating conditions, chains can carry high loads and satisfactorily withstand the shock of repeated reversals and, with proper lubrication, wear is reduced to a point where long life can be secured.

Belts

BELTED drives are probably the most common method of supplying rotative speed and transmitting power in machines. For use in designing rotative power transmission systems, an almost unlimited selection of leather, V and flat fabric belts is available. Generally speaking, leather belts are good to perhaps 7000 or 8000 revolutions per minute spindle or shaft speed and from 6000 to 8000 feet per minute belt speed. Standard V-belts should not be used where maximum lineal belt speed exceeds 5000 fpm or 6000 fpm at the most. Special V-belts, such as steel-core belts, will permit rotational speeds up to 12,000 rpm, with belt speeds running to 10,000 fpm. Finally, flat fabric belting, *Fig.* 14, can be operated at speeds as high as 20,000 fpm, with rotational speeds to 60,000 rpm or more. Indeed, woven belts have been run experimentally to 100,000 rpm on small grinding spindles—transmitted horsepower being low, of course.

Belt Speed and Pulley Diameter Critical

In designing a high-speed belt drive, the designer must, of course, be familiar with all the factors involved: Peak horsepower; amount and duration of overloads; belt speeds and pulley diameters available; contact angles; type of load; physical surroundings where drive will operate; positions of drive (horizontal, inclined or vertical); type of take up, if any; type of prime mover; installation requirements; etc. Of these factors, belt speed and pulley diameter are perhaps the most critical in high-speed belt drives. Increased minimum pulley diameter generally required as speed increases poses a definite limitation. Again as rotational speeds increase, the bearing problem inevitably becomes more troublesome. However, bearings giving satisfactory performance are available for present day commercial belt drives.

LEATHER BELTING: While generally used to drive relatively low-speed machinery requiring high horsepower, flat leather belt drives can, if properly designed for the job, be used at operating speeds to 8000 fpm. The American Leather Belting Association recommends a maximum peripheral speed of 6000 fpm for ordinary flat leather belt drives. Using this figure and assuming that a single-ply medium-weight leather belt runs at this speed on the minimum diameter pulley recommended (3 inches), a rotational speed of 7650 rpm is obtainable.

As in any other type of belt, centrifugal force is perhaps the major factor limiting speed of leather belts. As the chart in *Fig.* 15 shows, power transmission capacity levels off at about 6000 to 7000 fpm, dropping to zero at 10,000 to 12,000 fpm—depending on the coefficient of friction between pulley and belt on the particular installation. These curves are for double thickness belts with two values of initial tension and coefficient of friction. Tables are available from manufacturers giving power ratings for several weights of belt in various plies.

V-BELTS: As already mentioned standard V-belts should not be used at over 5000 fpm. However, steel-cable type V-belts can safely be used to peripheral speeds of 10,000 fpm and, with sub-standard sheave sizes, to as much as 12,000 rpm.

Centrifugal tension has always been considered the restraining factor in controlling the top speed of V-belts. Each particle of a belt, as it moves around a pulley, exerts a centrifugal force acting away from the pulley center. The sum total of these forces creates a tension which acts throughout the belt length.

There are two effects from centrifugal tension. One, increased tension in the belt sets up high stresses which result in heating and early fatigue failure. Two, increased tension tends to stretch the belt, lifting it from the pulleys and decreasing the avail-

Fig. 14—High-speed, endless flat belting on grinding spindle. Spindle and work rotate in opposite directions to get high grinder surface speed

able belt pull. Since centrifugal tension counteracts initial belt tension, at some high speed the two cancel each other out, making useful power transmission impossible.

Centrifugal tension increases as the square of the belt speed, or $T = WV^2/g$ where W = belt weight, lb per ft, V = belt speed, ft per sec and g = gravity acceleration, ft per sec.^{2} It is therefore obvious that high-speed belting should be made very light or relatively unstretchable, or both.

Several manufacturers are now producing V-belts in which the load is carried on high-tensile steel cables embedded in rubber. The tension member consists of a single cable coiled or looped a sufficient number of turns to give the desired strength and flexibility. The near-zero stretch and greater flexibility possible with this construction permits transmitting more than twice the horsepower of ordinary belts at linear operating speeds from 5000 to 10,000 fpm.

Design of V-Belt Drives: Recent experimental tests at Ohio State University* indicate that centrifugal force may have a lesser effect than has been commonly supposed. It is pointed out that the usual formula for computing centrifugal tension does not contain the radius of curvature of the belt, or the contact angle. Since the belt hangs between its pulleys in some sort of curve, this part of the belt also has a centrifugal tension WV^2/g which adds to the tension resulting from weight. This added tension between pulleys should exactly compensate for the centrifugal reduction at the pulleys. If the belt did not stretch the centrifugal force would have no influence on the transmissable power. Formulas to compensate for influence of centrifugal force should therefore contain terms reflecting the effect of tension on sag and belt length.

A greater effective pull can actually be transmitted at high speed (up to about 6000 fpm depending on belt section) than at low speed, as shown in *Fig.* 16, where T_1 is tight side tension and T_2 slack side tension. Tests plotted in *Fig.* 16 were made with smooth steel-cable *C*-section V-belts 96 inches long and with a locked center distance. *Fig.* 17 shows the difference between slip for cotton cord and steel-cable, notched V-belts on 9-inch pulleys, using a *C*-section belt 68 inches long.

The Ohio State University experiments emphasize, then, that as long as the belt *does not* stretch, the intimacy of contact between pulley and belt is not changed by

* C. A. Norman—"Belt Drives." MACHINE DESIGN, Feb. 1949, Page 111.

Table 3—Horsepower Rating per Belt for Steel Cable V-Belts

(Goodyear "Compass-V-Steel" belts—180° arc of contact)

A—Section Belts

Belt speed (fpm)	Pitch diameter of small pulley (inches)								
	2.0	2.4	2.8	3.2	3.6	4.0	4.4	4.8	5.0 (or over)
400	.21	.29	.34	.39	.43	.48	.53	.57	.60
1200	.61	.87	1.01	1.16	1.30	1.45	1.59	1.72	1.81
2000	.94	1.45	1.69	1.93	2.17	2.42	2.65	2.86	3.02
2800	1.16	2.03	2.37	2.70	3.04	3.39	3.71	4.00	4.23
3600	1.22	2.36	3.04	3.47	3.91	4.36	4.77	5.15	5.31
4400	1.08	2.47	3.46	4.21	4.77	5.25	5.63	5.94	6.09
5200	.69	2.34	3.51	4.39	5.07	5.62	6.07	6.44	6.60
6000	.01	1.91	3.26	4.27	5.06	5.70	6.21	6.64	6.83
6800		1.14	2.67	3.82	4.72	5.43	6.02	6.51	6.72
7600			1.70	2.99	3.99	4.79	5.44	5.99	6.23
8400			.31	1.73	2.83	3.72	4.44	5.04	5.31
9200					1.20	2.17	2.96	3.62	3.91
10000						.11	.97	1.68	2.00

B—Section Belts

Belt speed (fpm)	Pitch diameter of small pulley (inches)										
	3.0	3.4	3.8	4.2	4.6	5.0	5.4	5.8	6.2	6.6	7.0 (or over)
400	.25	.43	.53	.59	.65	.71	.76	.82	.88	.93	.98
1200	.73	1.25	1.59	1.77	1.94	2.12	2.27	2.46	2.63	2.80	2.95
2000	1.08	1.96	2.65	2.95	3.24	3.53	3.78	4.10	4.39	4.66	4.91
2800	1.25	2.48	3.45	4.13	4.54	4.94	5.29	5.74	6.15	6.52	6.87
3600	1.14	2.72	3.97	4.98	5.83	6.35	6.80	7.38	7.90	8.39	8.82
4400	.68	2.61	4.14	5.37	6.39	7.25	7.99	8.62	9.17	9.65	10.1
5200		2.08	3.89	5.35	6.55	7.56	8.43	9.17	9.82	10.4	10.9
6000		1.04	3.13	4.81	6.21	7.38	8.37	9.23	9.98	10.6	11.2
6800			1.79	3.70	5.27	6.60	7.72	8.70	9.55	10.3	11.0
7600				1.92	3.69	5.17	6.43	7.52	8.46	9.20	10.0
8400					1.36	3.00	4.33	5.59	6.64	7.56	8.38
9200						.02	1.54	2.86	4.01	5.01	5.91
10000									.49	1.57	2.55

C—Section Belts

Belt speed (fpm)	Pitch diameter of small pulley (inches)																	
	4.6	5.0	5.4	5.8	6.4	6.8	7.2	7.6	8.0	8.4	8.8	9.2	9.6	10.0	10.6	11.0	11.4	12 (or
400	.20	.45	.66	.84	1.02	1.09	1.15	1.21	1.28	1.34	1.40	1.47	1.54	1.60	1.71	1.77	1.83	1
1200	.54	1.28	1.91	2.46	3.05	3.26	3.46	3.64	3.83	4.01	4.21	4.40	4.61	4.80	5.12	5.32	5.48	5
2000	.67	1.90	2.96	3.87	5.02	5.44	5.76	6.06	6.39	6.68	7.02	7.34	7.69	8.00	8.53	8.86	9.14	9
2800	.45	2.18	3.66	4.93	6.54	7.46	8.06	8.48	8.95	9.35	9.83	10.3	10.8	11.2	11.9	12.4	12.8	13
3600		1.98	3.88	5.51	7.59	8.76	9.81	10.7	11.5	12.02	12.6	13.2	13.8	14.4	15.4	16.0	16.4	16
4400		1.15	3.48	5.48	8.01	9.45	10.7	11.9	12.9	13.84	14.7	15.5	16.2	16.8	17.7	18.1	18.7	19
5200			2.32	4.68	7.67	9.37	10.9	12.2	13.5	14.56	15.6	16.5	17.3	17.9	19.1	19.8	20.4	21
6000			.15	2.99	6.44	8.40	10.2	11.7	13.1	14.39	15.5	16.6	17.6	18.5	19.7	20.4	21.1	22
6800				.17	4.17	6.40	8.38	10.1	11.7	13.18	14.5	15.7	16.8	17.8	19.2	20.0	20.8	21
7600					.74	3.22	5.43	7.41	9.19	10.80	12.3	13.6	14.8	16.0	17.5	18.4	19.3	20
8400							1.18	3.37	5.33	7.12	8.73	10.2	11.6	12.8	14.6	15.5	16.5	17
9200										1.98	3.75	5.37	6.85	8.22	10.1	11.2	12.2	13
10000													.56	2.04	4.06	5.28	6.41	7

centrifugal tension. Pressure between pulley and belt will be determined by the applied initial tension. If the belt *does* stretch, initial tension will be influenced by the sag, and belt pull on the pulleys will be determined by the sag condition. On vertical drives, stretch caused by centrifugal tension may actually pull the belt free of the lower pulley. The experiments also showed that fixed center distance installations transmit greater effective pull than those with floating tensioning devices. The advantage of a floating drive with tensioning device lies in compensating for any permanent belt elongation.

For use on short center, small pulley, high-speed drives requiring high horsepowers, Goodyear has produced a notched, steel-cable V-belt. Notching increases belt flexibility, permitting use on smaller pulleys. The greater surface area and pumping action of the notches provide greater dissipation of heat. TABLE 3 gives horsepower ratings for *A, B* and *C* section Goodyear Compass V-Steel belts. Note the unshaded section giving belt ratings for *substandard* sizes. This section should be used with extreme caution. It is intended for use only where severe space limitations *dictate* substandard pulley sizes.

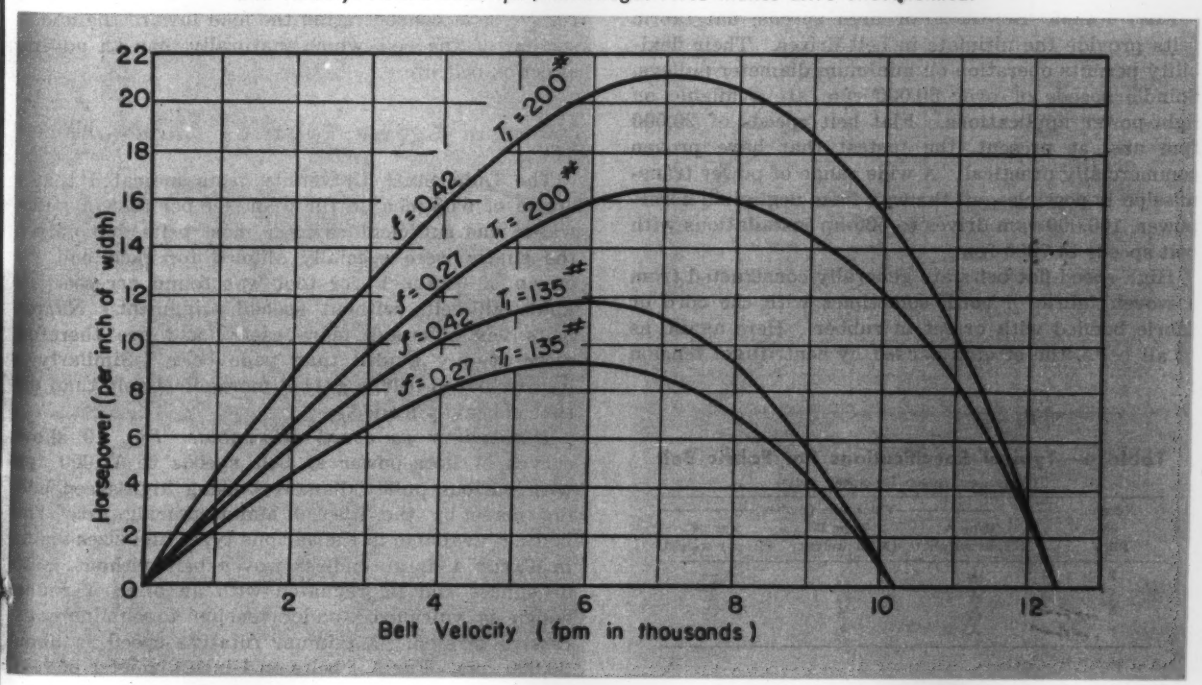
TABLE 3 values are for "light-duty" installations. *Light duty* is defined as intermittent service—not more than six hours per day intermittent operation and for loads never exceeding rated values. For other than light duty, correct table values as follows: For *normal duty*—i.e., occasional starting or peak loads to 150 per cent of full load or continuous service from 6 to 16 hours per day—multiply the required hp load by 1.2. For *heavy-duty* jobs—where occasional starting or peak loads do not exceed 200 per cent of full load, or for continuous service from 16 to 24 hours per day—multiply hp rating by 1.4. For *extra-heavy duty*—over 200 per cent of full load occasionally on starting or under peak-load conditions, where starting or peak loads and overloads are frequent, or for continuous service 24 hours per day, 7 days a week—multiply required horsepower rating by 2.0 before picking a sheave size from the table.

The upper limit of 10,000 fpm given in these tables for steel-cable V-belts is not determined so much by centrifugal tension and resultant belt stretch as it is by unbalance of the sheave. At high speeds, vibrations tend to "tip" the belt and cause it to roll out of the pulley groove. Therefore, for belt speeds over 5000 fpm, specially balanced pulleys are a "must." Otherwise trouble will be encountered from belt instability due to vibration.

Applications: Fan and generator drives on some cars make use of steel-cable V-belts. Drive conditions are severe—belt speeds to 8000 fpm when car speed is around 80 mph, small pulley diameters, small contact angles, and high loads at low speeds. This type belt has worked out well in these installations, however.

The Merco Centrifugal Co. uses Goodyear Compass V-Steel belts to drive centrifuges at high speed and under high loads. One model employs a 50-hp motor driving a rotor at 6000 rpm through 10 B-section belts traveling at 8800 fpm. Another Merco Centrifuge, *Fig.* 18, is driven by a 100-hp motor through 10 C-section belts to give a rotor speed of 2900 rpm at 7750 fpm belt speed. In *Fig.* 18, the lower shaft

Fig. 15—Relationship between carrying capacity of double leather belts and velocity. Above 6000 fpm, centrifugal force makes drive uneconomical



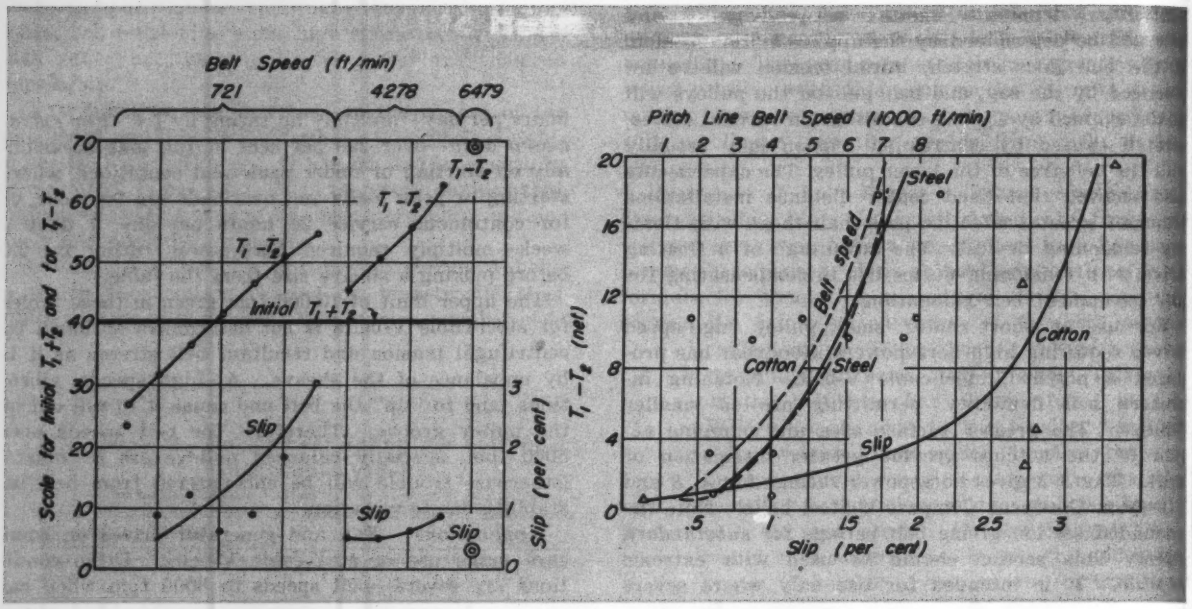


Fig. 16—Comparative speed tests with C-96 smooth steel-cable V-belt on 9-inch pulleys with locked center distance

Fig. 17—Comparative high-speed tests using C-68 notch belts on 9-inch pulleys

and rotor are not yet in position. A smaller Merco centrifuge, in which the drive consists of a 15-hp motor connected through five A-section belts to the 9000-rpm spindle, has a belt speed of 7300 fpm.

In all three of these centrifuges, the motor is mounted on a bracket supported on a roller bearing track. Required running tension is applied to the belts through a compression spring. The motor is free to move laterally a short distance against spring compression to compensate for sudden changes in load, as during starting.

FLAT FABRIC BELTS: For high speeds, flat fabric belts provide the ultimate in belt drives. Their flexibility permits operation on minimum diameter pulleys. Spindle speeds of over 60,000 rpm are available on light-power applications. Flat belt speeds of 20,000 fpm are, at present, the fastest that have proven commercially practical. A wide range of power transmission is possible—all the way from negligible horsepower, 100,000-rpm drives to 500-hp installations with belt speeds to 7000 fpm.

High-speed flat belts are generally constructed from a woven fabric or cord, sometimes with the cord or fabric bonded with or set in rubber. Here again, as in all belts, the stretch caused by centrifugal tension results in decreased power capacity, and since centrifugal force depends on the belt mass, most high-speed belting is made as light as possible consistent with the tensile strength required. Belts are made endless and usually molded or formed under tension to eliminate excessive stretch after being put into service.

Shaft alignment is critical in high-speed installations and should be carefully specified. The least possible pulley crown that will keep the belt from running off should be used. Excessive crowning has the effect of concentrating the load toward the middle section of the belt which, naturally, has an adverse effect on belt life.

Table 4—Typical Specifications For Fabric Belt
(Based on Rusco Hy-Speed belt)

Number of Plies	Thickness (inches) Type A (Light weight)	Type B (Med. weight)	Type C (Heavy weight)
2	1/32	3/32	—
4	1/16	1/32	3/32
6	3/32	1/16	1/32
8	1/16	3/32	1/16

Minimum Crown Requires Alignment

The Ohio State University tests indicated that a crown of 0.06-inch on the diameter per foot of pulley width was sufficient to steer most belts, but only if the shafts were specially aligned for each belt. A crown of 0.10-inch per foot was found necessary to guide all belts without special alignment. Narrow belts flex sideways more easily, and are therefore more readily guided than wide belts. Similarly, a lightly loaded belt is guided more effectively than one that is heavily loaded.

Horsepower vs speed attainable: *Fig.* 19 shows curves of horsepower at belt speeds to 12,000 fpm with various pulley diameters using high-speed belting made by the Russell Manufacturing Co. This belting, available in the various types and sizes shown in TABLE 4, is an endless woven belt without joints or splices and impregnated with an oilproof rubber which is vulcanized under tension to eliminate excessive stretch. Maximum rotative speed is about 30,000 rpm. For A-4 belts on 1-inch diameter pulleys,

rpm = 8000 × 12 ÷ 3.14 × 1 = 30,600. Belts are made in widths from 3/16-inch to 10 inches and in 2, 4, 6, and 8 plies. These belts have been applied to internal grinders, surface grinders, tool grinders, drill presses, carvers, turning lathes, etc., *Figs.* 20, 21 and 22.

For some extremely high-speed special applications, when the belt operates on pulleys one inch or less in diameter, the Russell Manufacturing Co. recommends an endless hand-woven belt in either straight or twill weaves and with various treatments or impregnation depending on the application. These belts are giving highly satisfactory results on internal grinder drives, among others, where belt speeds exceed 18,000 fpm (speed of a spindle with a 1-inch pulley = 18,000 × 12 ÷ π × 1.0 = 68,500 rpm). These belts are available in widths from ¾ to 3 inches.

Onsrud Machine Works Inc. uses a vacuum pulley, *Fig.* 23, to prevent belt slippage and permit high spindle speeds. Using an endless, woven, rubber-impregnated belt, this drive has been used to transmit up to 15 hp at speeds to 20,000 rpm in woodworking machinery. In some grinding machines and other special equipment, this pulley transmits light loads to 30,000 rpm. The maximum spindle rotation possible with the vacuum pulley is about 40,000 rpm, the limiting factor being the surface speed of the belt. Onsrud uses special fans and compressed air to dissipate heat resulting from bearing and belt friction. A cross-section view of an automatic shaper spindle assembly is shown in *Fig.* 24. This application provides a spindle speed of 10,000 rpm.

Flat woven belts have been used extensively by the Sharples Corp. to drive centrifuge rotors at speeds ranging to 15,000 rpm. *Fig.* 25 shows a machine employing a quarter-turn drive in which tension is maintained by a counterweighted idler. This installation is powered by a 2-hp, 3450-rpm motor using a 1-inch woven belt traveling at about 9000 fpm over 10 and 2¼-inch diameter pulleys. The drive pulley is, in this case, made of Bakelite, with a suitable filler material to give a high coefficient of traction friction. The spindle pulley is steel and is constructed as shown in *Fig.* 26. Pulley crowns are slight since high crowns are not necessary at high belt speeds

Fig. 18—Merco Centrifugal Co. Model B-30 centrifuge powered by 100-hp motor through 10 C-section Compass V-steel belts. Shaft speed, 2900 rpm, belt, 7750 fpm

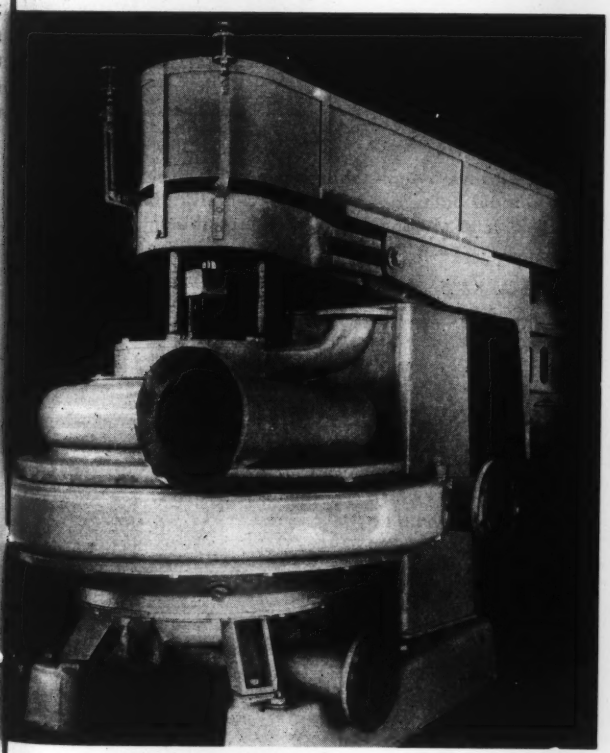
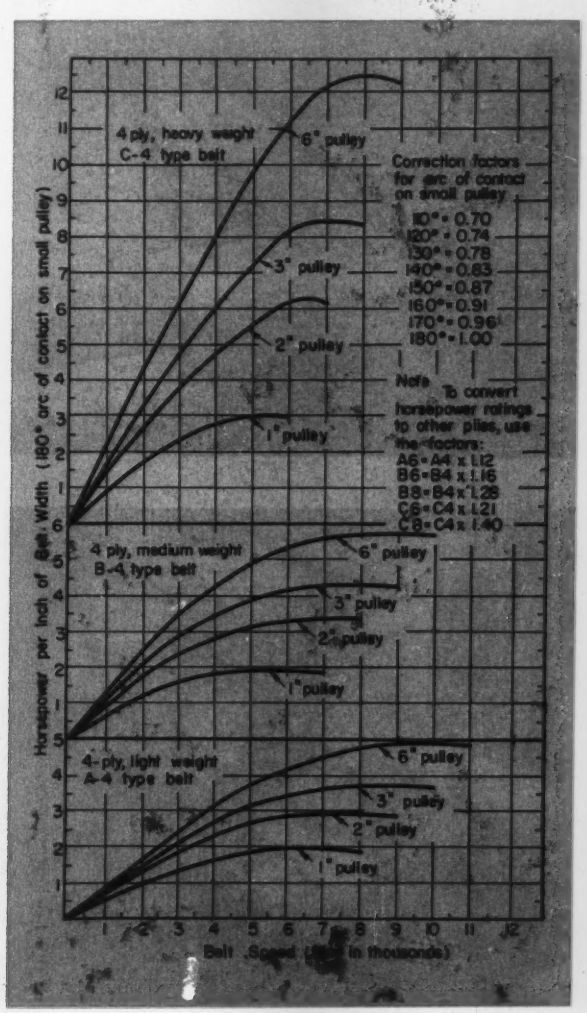


Fig. 19—Belt horsepower versus belt speed for Rusco Hy-Speed endless woven belts. Curves are for 4-ply, A, B and C section belts with 1, 2, 3 and 6-inch pulleys. Values from table should be corrected for other plies or pulley contact angle other than 180 degrees. Belts are available in widths from 3/16 to 10 inches



and, as mentioned previously, actually tend to split the belt.

DESIGN OF FABRIC BELT DRIVES: A series of interesting charts and tables have been compiled by The Gates Rubber Co. to aid designers of high-speed machinery. Data for designing flat-belt drives for high-speed machinery are based on the fact that a belt, as it operates around the wheels of the drive, is subject to a cyclic stress pattern characterized by one stress peak for each wheel of the drive. It has been found that the arithmetic mean of all the stress peaks is logarithmically related to the total number of stress cycles which the belt can endure before failure in operation.

Fig. 27 shows how the stresses to which the belt is subject as it operates on the drive combine in such a way as to produce a stress peak at each wheel. For purposes of illustration, consider the two-wheel drive shown in the upper right-hand corner. Since the left-hand wheel is the driver and the direction of rotation is clockwise, the strand of belt from C to D will be under more tension than the strand running from E to B. Tension in the belt strand CD is denoted by T_1; that in the strand EB by T_2.

In addition to the stresses T_1 and T_2 two other stresses are imposed on the belt. Since the belt travels a curved path it will be subject to a centrifugal stress T_C. As it bends over a wheel, the belt will also be subject to a bending stress T_B whose magnitude will be inversely proportional to the diameter of the wheel over which it is running.

The heavy line in the *Fig.* 27 shows how these stresses combine at various points on the drive. As indicated by this line, at the point A the belt is subject to centrifugal stress T_C and to the loose side tension T_2. Then, as the belt is bent at the point B to the diameter of the driven wheel, it is immediately subject to the bending stress T_{BN}. The rise in the stress curve from point B to point C represents a rise in the external tension from T_2 at point B to T_1 at point C. As the belt straightens out at point C, the stress due to bending is decreased by the amount T_{BN} and then remains constant at the level $T_1 + T_C$ until it bends over the driver wheel at point D.

Here the reverse of the process on the driven wheel takes place. The bending stress, T_{BR}, however, is

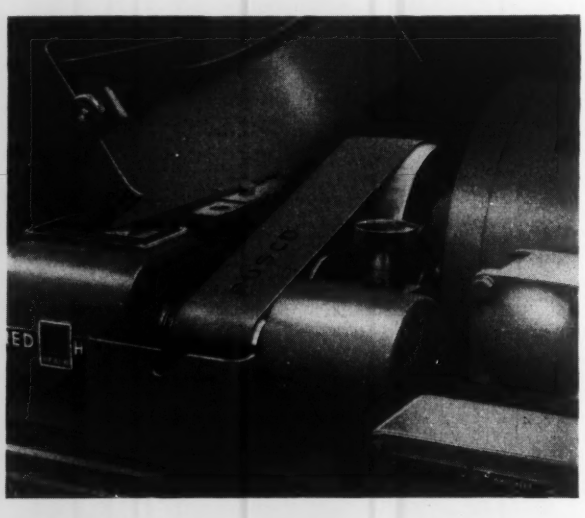


Fig. 20—Left—Rusco flat belting applied to Heald Model 181 high-speed internal grinder. Note idler on outside of lower strand of belt

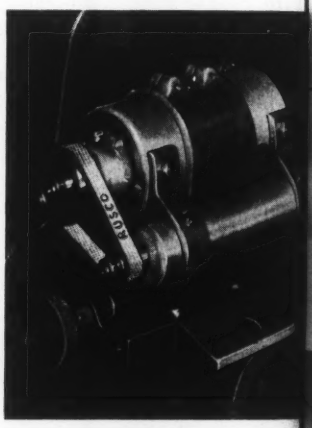


Fig. 21 — Left, below—Hy-Speed endless woven belting used on spindle carver

Fig. 22—Above, right—Short-center, high-speed grinder drive without idler or adjustable centers. Endless belts are vulcanized under tension to reduce stretch in application where take-up is a problem

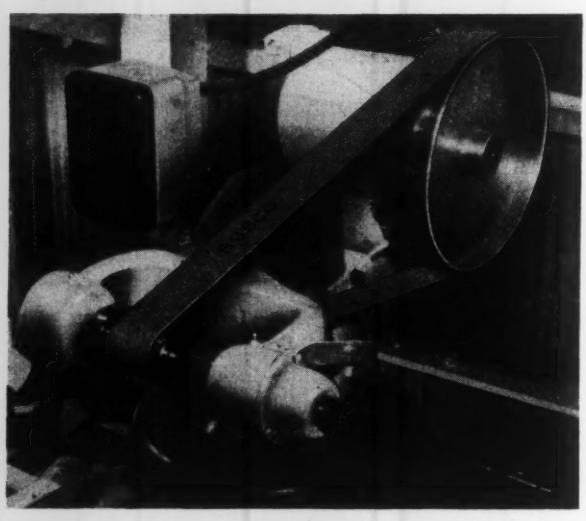


Fig. 23—Right—Onsrud vacuum pulley which permits top spindle speeds of 40,000 rpm without excessive slip

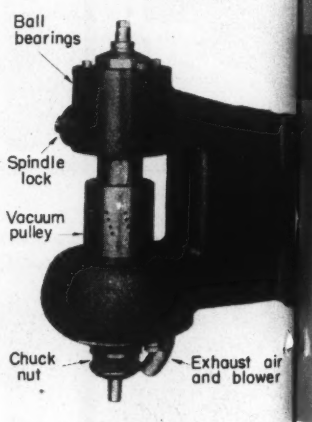


Table 5—Service Factors for High-Speed Machines*

1.0	1.3
Blowers	Grinders
Centrifugal superchargers	Positive displacement blowers
Centrifugal pumps	Woodworking tools:
Centrifuges	Multiple carvers
Other machines with smooth load or no overload	Routers
	Shapers
	Other machines with pulsating load or infrequent short duration overload

*Courtesy Gates Rubber Co.

Table 6—Acceptable Index for High-Speed Belt-Driven Machines†

Machine	Number of Wheels 2	3	Index *
Centrifugal superchargers	2	3	1
Centrifuges, continuous service	50	75	25
Intermittent service	8	12	4
Grinders, Production	20	30	10
Tool post	10	15	5
Home workshop	2	3	1
Multiple carvers, production	12	18	6
Home workshop	2	3	1
Routers, production	12	18	6
Home workshop	2	3	1
Shapers, production	12	18	6
Home workshop	2	3	1

† Courtesy Gates Rubbers Co.

* If a drive has more than 3 wheels, compute the acceptable index: Multiply the number of wheels by the number in the index column. The result is the acceptable index of the drive.

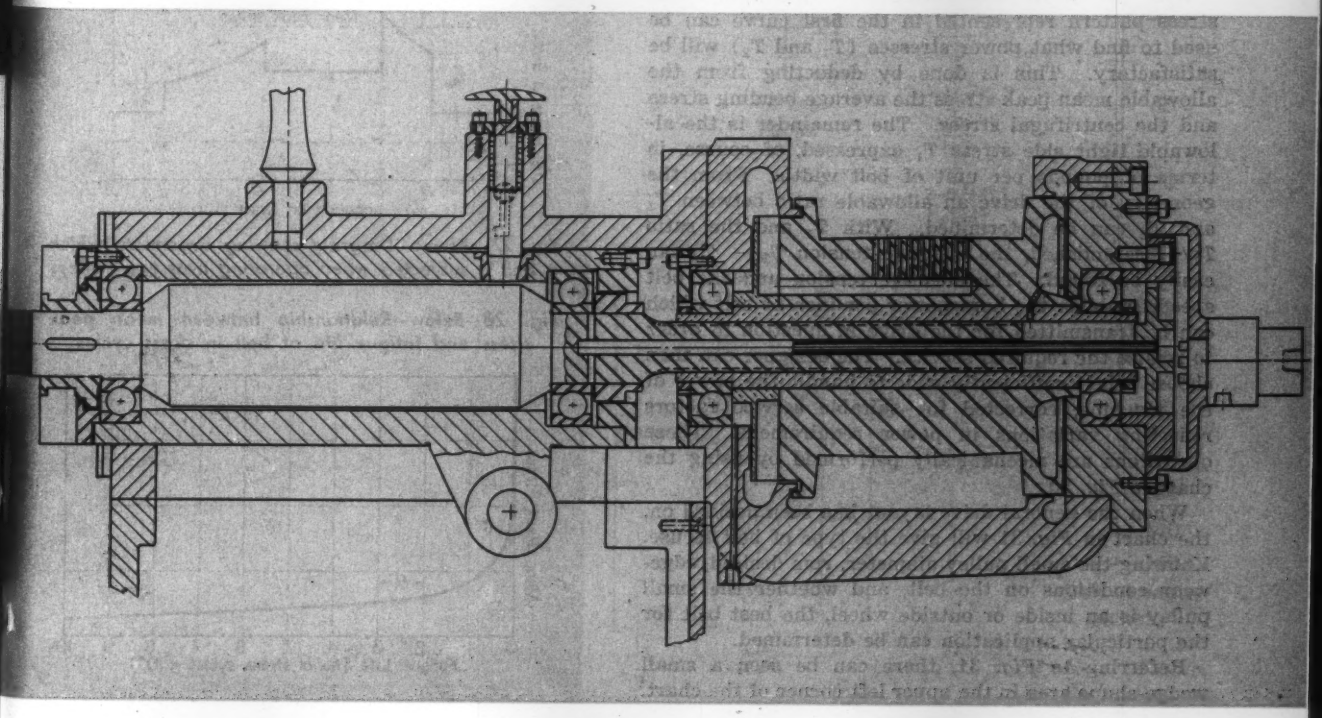
smaller than the bending stress T_{BN}, since the diameter of the driver is larger than that of the driven wheel. The external tension decreases between points D and E from an amount T_1 to T_2. Finally, as the belt leaves point E, the stress decreases by the amount T_{BR} and remains constant until it has completed a revolution at point A.

Inspection of *Fig.* 27, then, shows that the belt is subject to a peak stress each time it bends over a wheel. Each of these peaks is termed a stress cycle. It has been found that the arithmetic mean peak stress level indicated by the dotted line is an acceptable criterion to use in establishing the fatigue life of the belt.

The curve in *Fig.* 28 shows the relationship between the mean peak stress and the fatigue life of the belt expressed in hundreds of millions of stress cycles. Three regions have been marked on the graph. The region A, from 8 to 9 hundred million stress cycles, represents a fatigue life which typical high-speed production applications must have for satisfactory drive service. The region B, from 2 to 3 hundred million stress cycles, represents a satisfactory fatigue life for high-speed, intermittently used machinery. Point C, 10 million stress cycles, corresponds to satisfactory service life for a highly specialized, very intermittent, extremely short-time service application for flat fabric belt drives.

Given the belt length, belt speed, the number of wheels on the drive, and the time-service required for

Fig. 24—Cross section of Onsrud vacuum pulley assembly used on belt drive of automatic shaper. Spindle speed in this case is 10,000 rpm



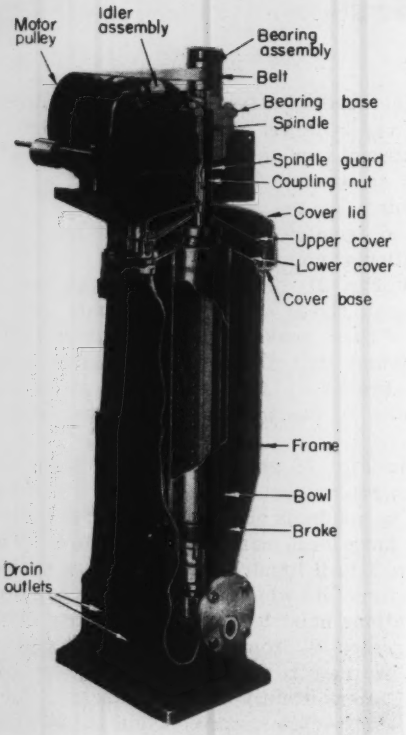


Fig. 25—Left—Sharples Vaportite Centrifuge showing quarter-turn flat belt drive to rotor spindle. Spindle rotates at 15,000 rpm with 9000-fpm belt speed

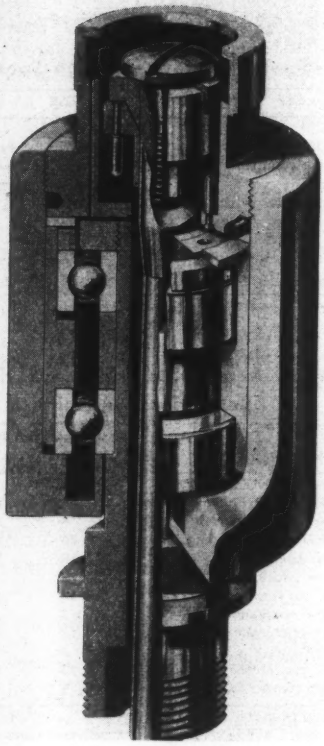


Fig. 26 — Right — Cutaway drawing of bearing and pulley assembly in Sharples centrifuge, Fig. 25. Diameter of this steel pulley is 2¼-inches

satisfactory drive operation, the total number of stress cycles which the belt must endure before failure can be determined. Then, from the curve, the acceptable mean peak stress to insure the required fatigue life can be determined. These operations are given in chart form in *Fig.* 29.

With the mean peak stress level determined, the stress pattern represented in the first curve can be used to find what power stresses (T_1 and T_2) will be satisfactory. This is done by deducting from the allowable mean peak stress the average bending stress and the centrifugal stress. The remainder is the allowable tight side stress T_1 expressed, of course, in terms of pounds per unit of belt width. From the geometry of the drive an allowable ratio between T_1 and T_2 can be determined. With T_1 and the ratio T_1/T_2 established, the effective tension $T_1 - T_2$ can easily be found. The effective tension and the belt speed establish the horsepower per inch of belt which can be transmitted for the drive in question in order to realize the required service. The designer can then choose a belt sufficiently wide to handle the power of the machine, corrected for suitable service factors reflecting variations in power requirement. These operations are mechanically performed by using the chart in *Fig.* 30.

When the general drive layout has been decided on, the chart in *Fig.* 31 will give the type of belt to use. Knowing the small pulley diameter, rpm desired, edge-wear conditions on the belt, and whether the small pulley is an inside or outside wheel, the best belt for the particular application can be determined.

Referring to *Fig.* 31, there can be seen a small wedge-shape area in the upper left corner of the chart.

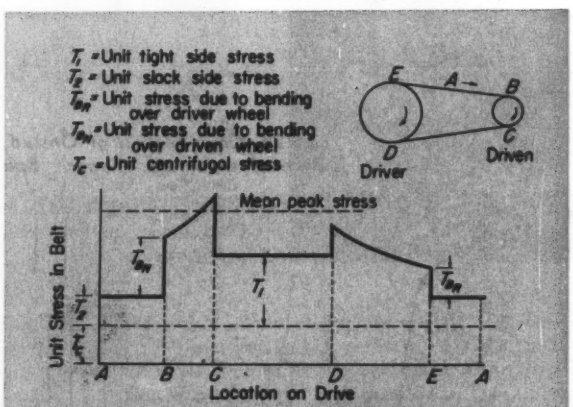


Fig. 27—Above—Diagram showing stresses to which a belt is subjected when operating over two pulleys

Fig. 28—Below—Relationship between mean peak stress and fatigue life of belt in stress cycles

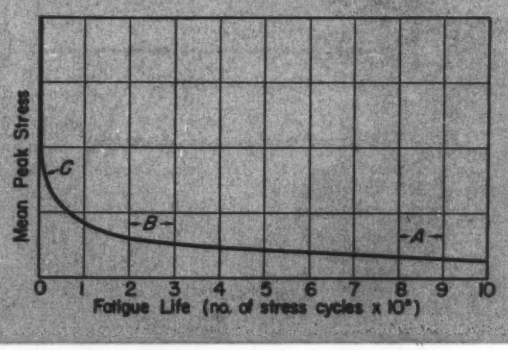


Fig. 29—Chart for determining the acceptable mean peak stress level (or stress factor) of a belt to give satisfactory performance and operating belt life

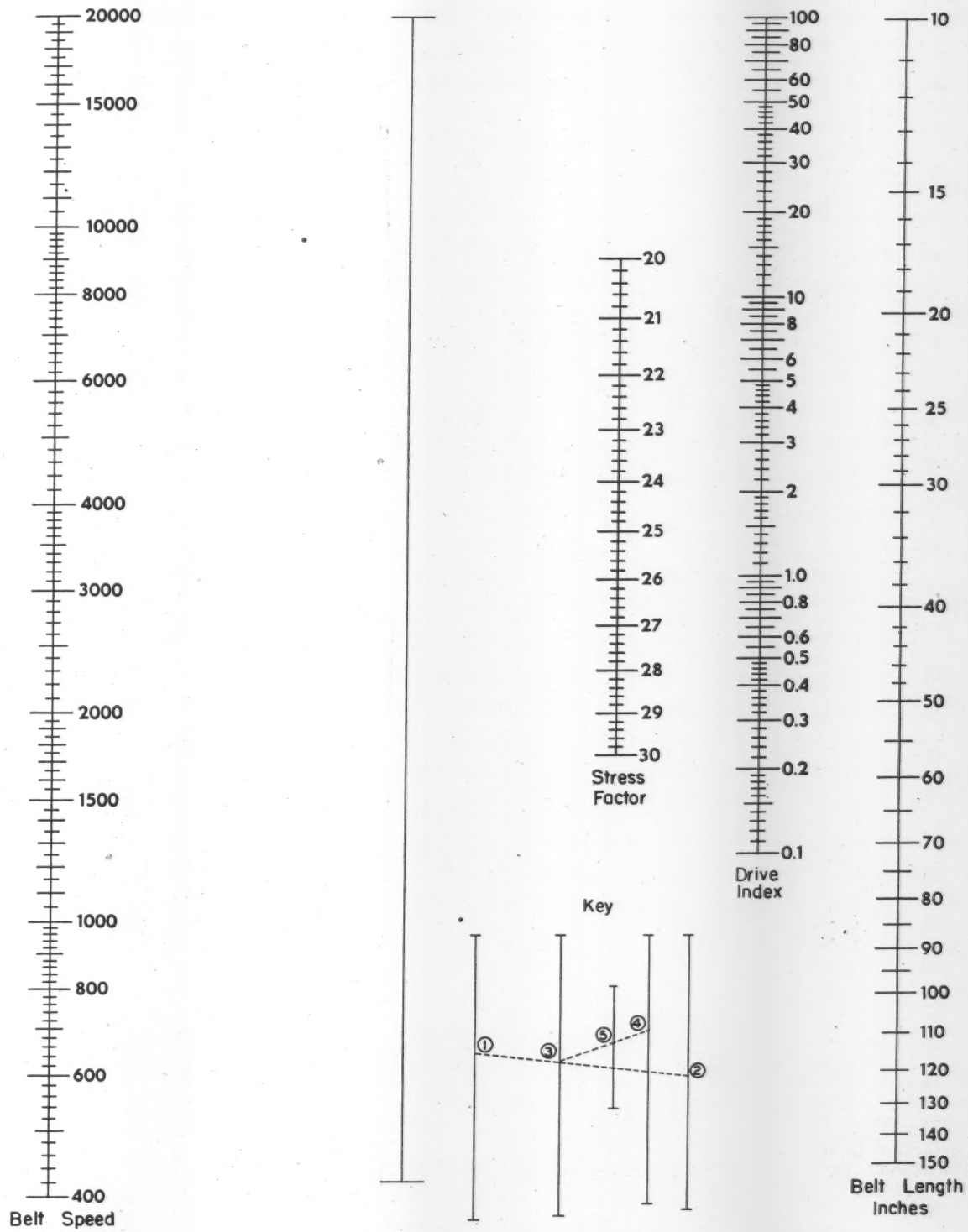
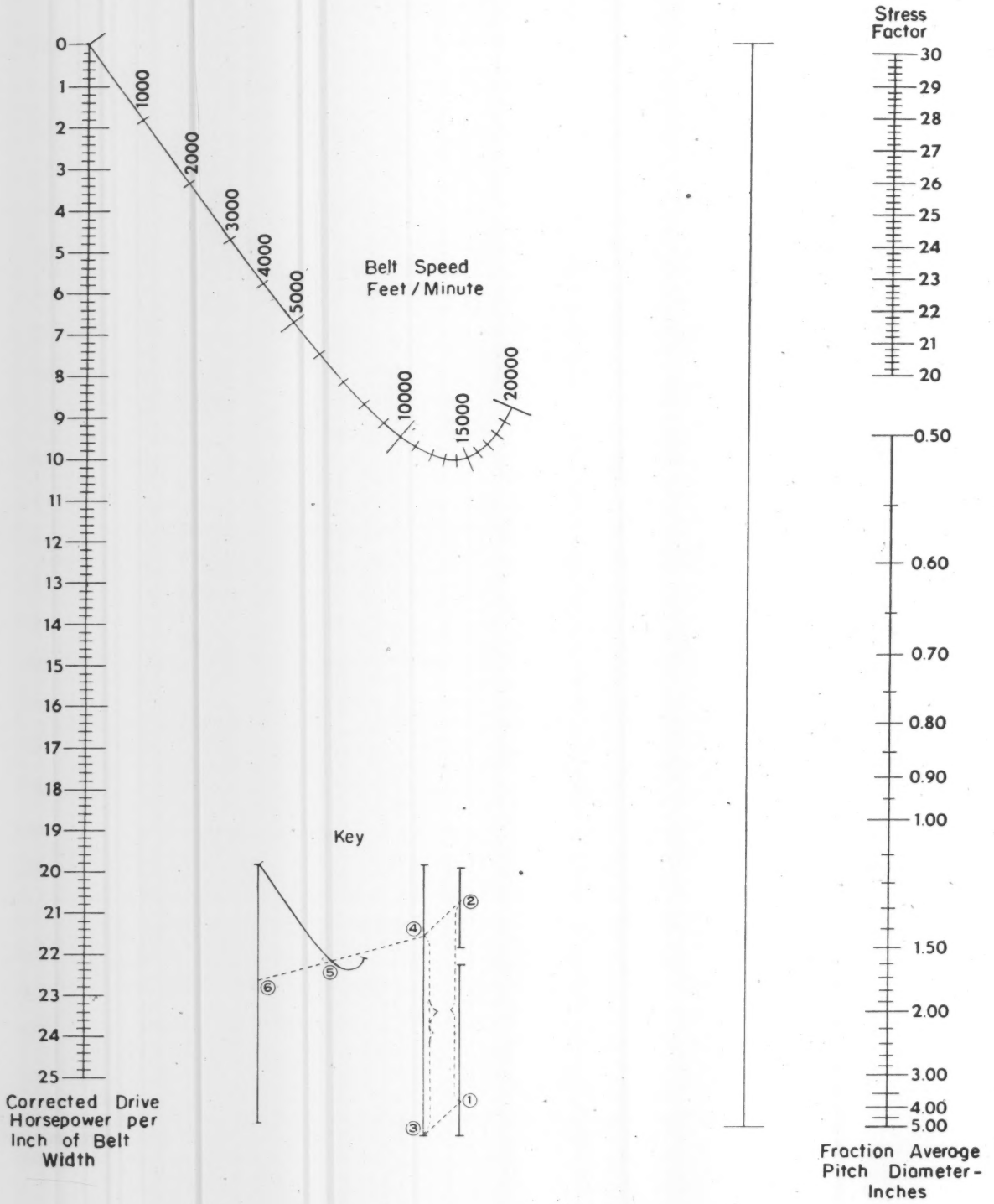


Fig. 30—Chart for finding safe horsepower per inch of belt width when fraction average pitch diameter, stress factor and belt speed are known. From the key it can be seen that construction line from the stress factor is made parallel to that from the pitch diameter drawn to the index



The upper line of this wedge represents a constant belt speed of 20,000 fpm; the lower curved line, partially dotted, represents a line of constant centrifugal force. Inside that particular area, a Speed-Cord or Speed-Flex belt running over an outside wheel might develop centrifugal forces large enough to tear the cord cover from the belt. If it is necessary for a drive to operate within this area, pulleys must be on the inside of the belt only. The Gates people have designed Speed-Flex drives for spindle speeds of 100,000 rpm. Although of a highly experimental nature, these drives have been successful—but little power was involved.

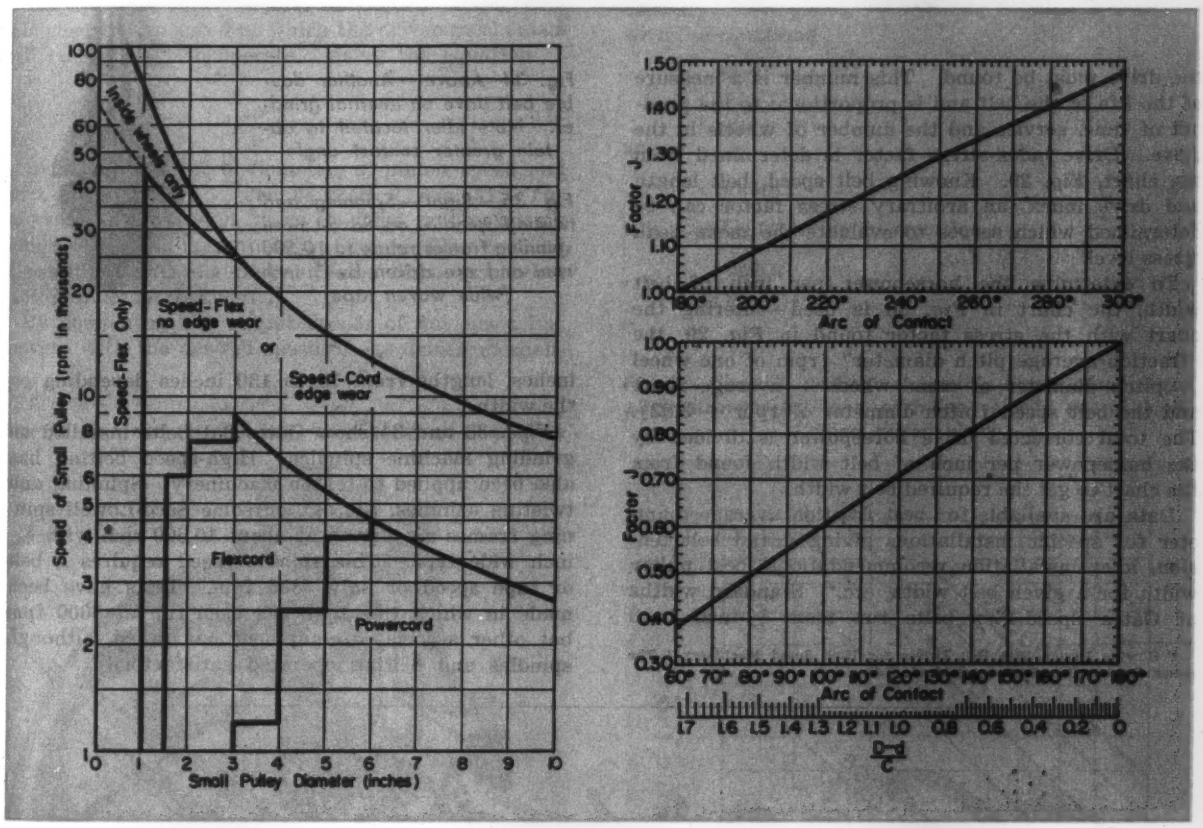
HIGH SPEEDS

When a belt has been decided on, and horsepower requirements settled, the service factor should be selected. TABLE 5 gives acceptable service factors for use with representative high-speed machines. Next, the design horsepower should be corrected for arc of contact by finding the J factor from *Fig.* 32, where D and d are pulley diameters and C is the minimum center distance, all in inches. Corrected drive horsepower is design horsepower $\times$ service factor $\div J$.

Next, from TABLE 6 the acceptable drive index for

Fig. 31—Selection of the best belt for a drive when small pulley diameter and rpm are known, can be made from chart based on maximum belt speed of 20,000 fpm

Fig. 32—Chart for obtaining J factor used in correcting horsepower for arc of contact other than 180 degrees



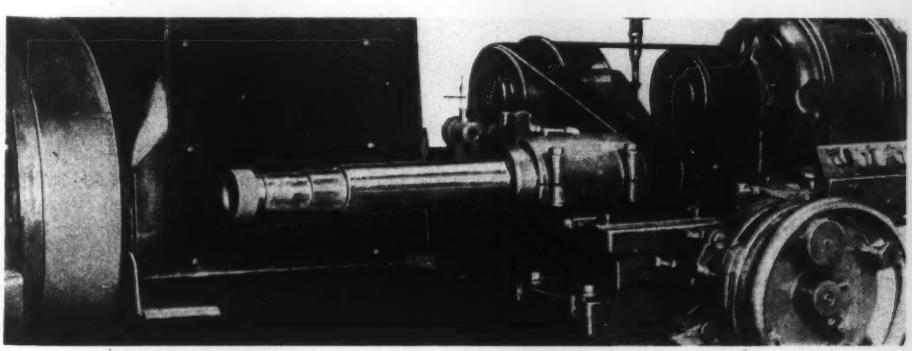
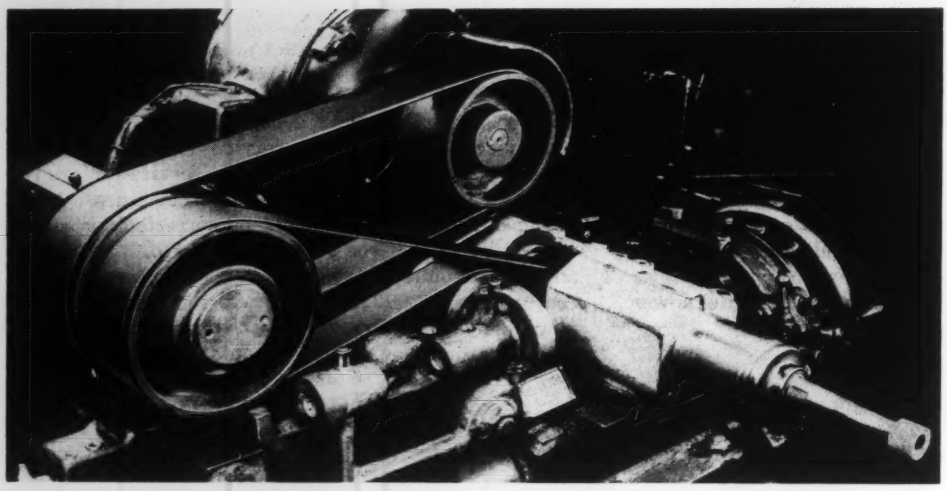


Fig. 33 — Left — Two Gates flat belts used on high speed internal grinding spindle



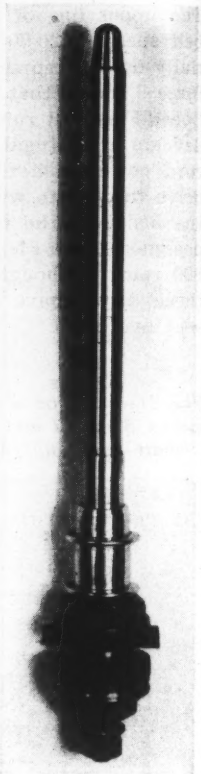


Fig. 34—Above — Another double belt drive on internal grinder. Note idler located to obtain greater contact angle

Fig. 35 — Right — Spinning and twisting spindles on Saco-Lowell spinning frames rotate to 10,500 rpm and are driven by ¾-inch wide woven tape

the drive must be found. This number is a measure of the life of the belt and is proportional to the product of time, service and the number of wheels in the drive. Drive index-stress factor is determined from the chart, *Fig.* 29. Knowing belt speed, belt length and drive index, an arbitrary stress factor can be determined which serves to evaluate the mean peak stress level.

To determine the horsepower per inch of belt width, the chart in *Fig.* 30 is used—entering the chart with the stress factor found in *Fig.* 29, the "fraction average pitch diameter" (rpm of one wheel × pitch diameter of same wheel ÷ average rpm) and the belt speed (pitch diameter × rpm ÷ 3.82). The total corrected drive horsepower is divided by the horsepower per inch of belt width found from the chart to get the required belt width.

Data are available for best fraction average diameter for specific installations giving initial belt tension, idler installation recommendations, best pulley width for a given belt width, etc.* Standard widths of Gates Speed-Flex belts run from ½-inch to 3 inches, lengths from 15 to 130 inches depending on the width.

Figs. 33 and 34 show Gates flat belts installed on grinding machine spindles. High-speed belting has also been applied to textile machinery. Spinning and twisting spindles, *Fig.* 35, on some Saco-Lowell spinning frames are driven at about 10,500 rpm with ¾-inch wide tape. This spindle speed requires a belt or tape speed of only 3855 fpm. Tests have been made in which this tape has been run at 8000 fpm but other associated equipment has failed, although spindles and belting operated satisfactorily.

*"How to Design Flat Belt Drives for High Speed Machinery," The Gates Rubber Co., Denver, Colorado.

Gears

INTRODUCTION of higher speeds for prime movers has prompted the development of high-speed gearing to connect the prime movers to driven mechanisms such as blowers, compressors, and other machinery. Less than a decade ago, the demand for this type of gearing became such that the manufacturers of high-speed drives felt the necessity of assembling their knowledge, experience and test data into tentative standards. These in turn were modified and in 1947 were published as AGMA Standard Practice 421.02, "High Speed Helical and Herringbone Gear Units."*

Essentially, high-speed gearing comprises a totally-

* "New High-Speed Helical and Herringbone Gear Standard"—Data Sheet, MACHINE DESIGN, Dec. 1947, Page 157.

enclosed drive including gears, bearings, oil seals, breather, gear case lubricating and cooling systems, and related auxiliary equipment, *Fig.* 36. Each of these components has a definite effect upon drive efficiency and imposes limitations relative to high-speed. It is the purpose of this discussion to outline the general factors of gear and tooth sizes, ratios, tooth profile accuracy, etc., which permit effective operation at high speeds. High-speed gearing to be covered will be specifically that having speeds of 4000 revolutions per minute and higher and pitch line velocities of 4000 feet per minute and higher.

SELECTION OF GEARING: It is most feasible to present high-speed gear limitations by developing a series of established equations in a logical design order. Prime mover data are usually known; the mechanism to be driven is given specified running conditions. The problem is to design a drive to link the two.

The speed at which the pinion has to revolve is established by the speed at which the driven mechanism will be expected to operate. Using the equation

$$V = \frac{\pi d}{12} n \quad \ldots\ldots (1)$$

a chart relating pitch diameter, d, pinion speed and pitch-line velocity, V, can be plotted, *Fig.* 37. The information from *Fig.* 37 is of value in illustrating pinion size variation with respect to speed but to make it useful in selecting proper pinion diameters, other factors must be considered.

By comparing the output speeds of various prime movers with the desired speed of the driven mechanism, it is possible to establish the gear ratio needed. A study of center distance limitations must also be made. These three factors, ratio, R, center distance, C, and pinion diameter, d, are interdependent and can be determined by using the equation:

$$C = \frac{d\,(R+1)}{2} \quad \ldots\ldots (2)$$

The foregoing equation is plotted at the lower right of *Fig.* 38. Used in conjunction, *Figs.* 37 and 38 give a general idea of proportions for gears capable of high speeds but, from the standpoint of speed only.

Practicability of the sizes selected from *Figs.* 37 and 38 can be determined by considering the power capacity of the gears. Combining Equations 1 and 2 with the equations

$$P = \frac{WFV}{33{,}000} \quad \ldots\ldots (3)$$

and

$$P = C^2FnMK \quad \ldots\ldots (4)$$

it is possible to derive the equation

$$W = \frac{2CKR}{(R+1)^2} \quad \ldots\ldots (5)$$

Fig. 36—Cutaway view of a typical high-speed gear drive

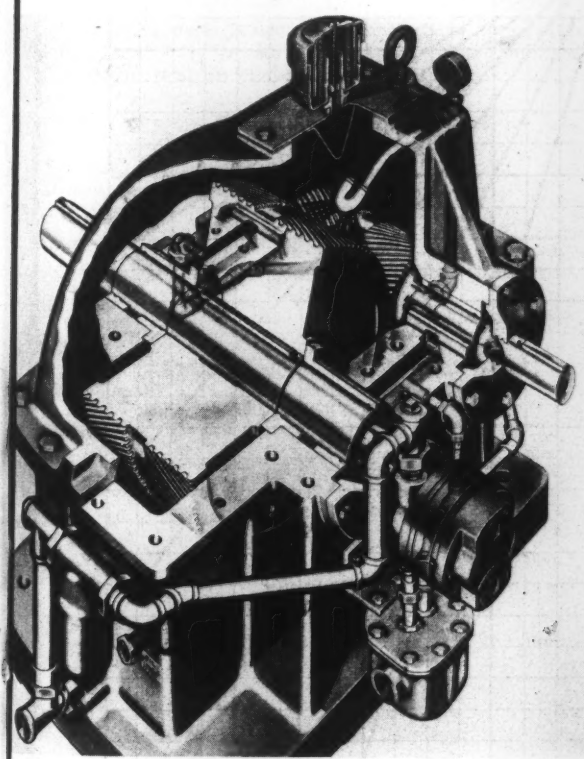
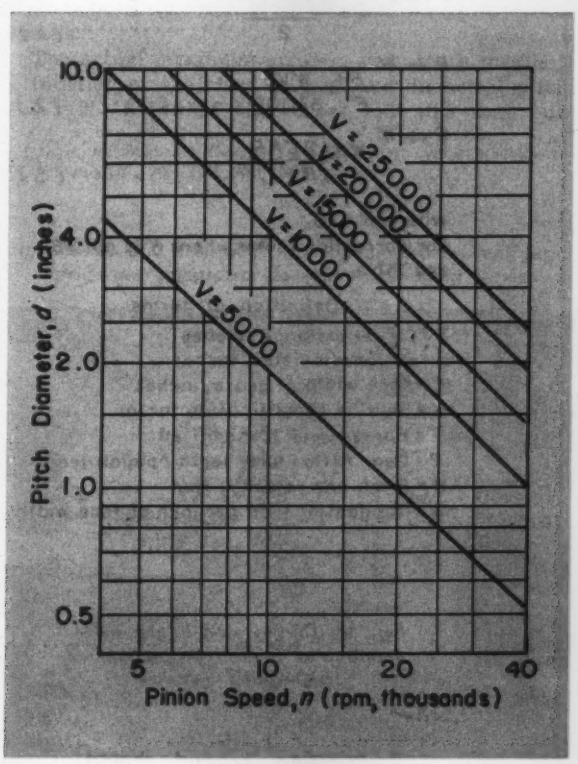
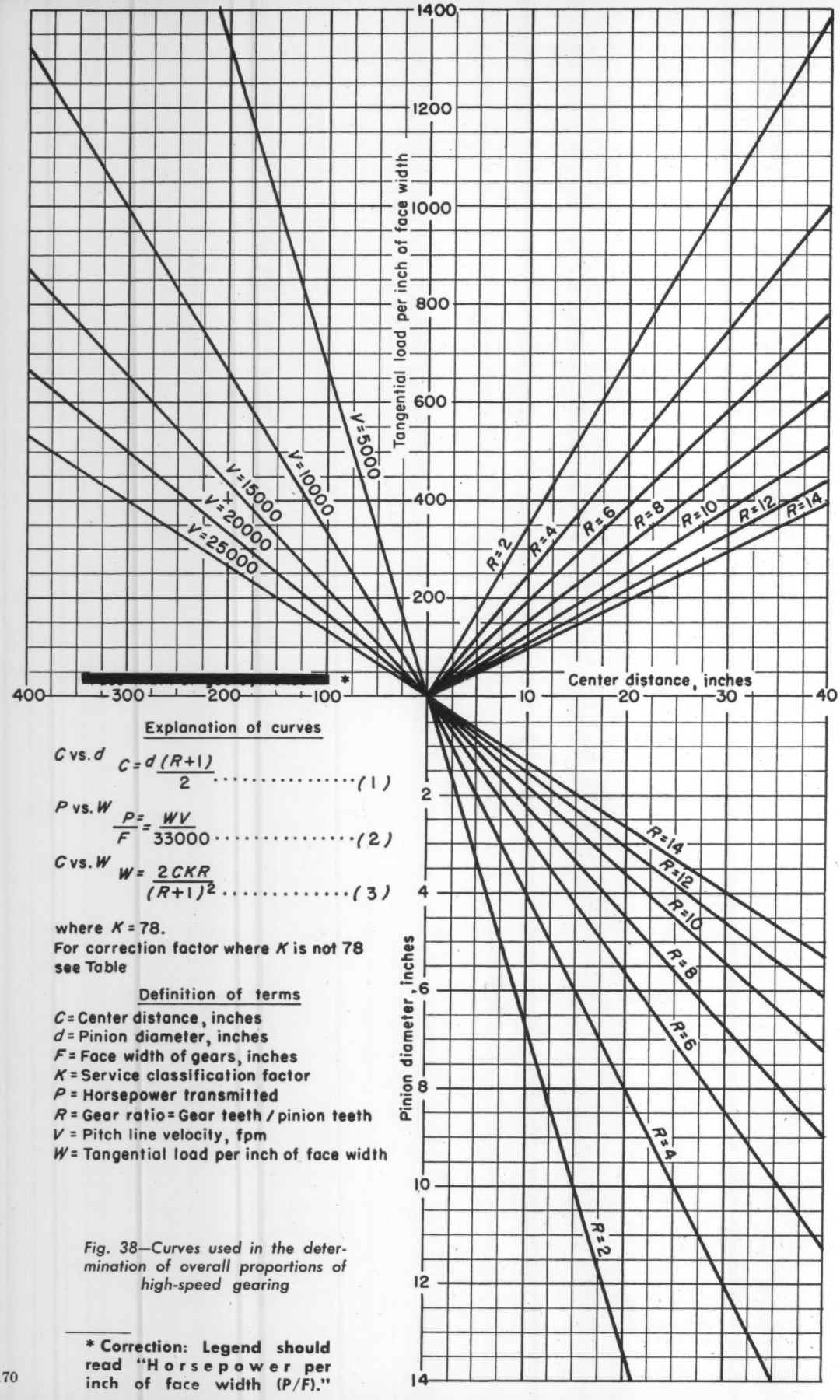


Fig. 37—Chart illustrating relationship between speed (n) pitch diameter (d), and pitch-line velocity (V) of high-speed driven pinion





Explanation of curves

C vs. d $C = \frac{d(R+1)}{2}$ ..... (1)

P vs. W $\frac{P}{F} = \frac{WV}{33000}$ ..... (2)

C vs. W $W = \frac{2CKR}{(R+1)^2}$ ..... (3)

where $K = 78$.

For correction factor where K is not 78 see Table

Definition of terms

C = Center distance, inches
d = Pinion diameter, inches
F = Face width of gears, inches
K = Service classification factor
P = Horsepower transmitted
R = Gear ratio = Gear teeth / pinion teeth
V = Pitch line velocity, fpm
W = Tangential load per inch of face width

Fig. 38—Curves used in the determination of overall proportions of high-speed gearing

* Correction: Legend should read "Horsepower per inch of face width (P/F)."

which establishes a relationship between load transmitted per inch of face width, *W*, center distance, *C*, and gear ratio, *R*. This is plotted at the upper right of *Fig.* 38.

The service classification factor, *K*, has varying values based on the maximum continuous load to be transmitted and has been tabulated in AGMA Standard Practice 421.02. For the purpose of constructing the chart, an average value of 78 was assigned to *K*. Where the applications require *K* factors other than 78, the load per inch of face width, *W*, should be multiplied by the correction factor indicated in TABLE 7.

All that remains to complete these charts is to convert the load transmitted per inch of face width into terms of horsepower transmitted per inch of face width, *P/F*. This can be accomplished by using Equation 3. The relationship is plotted at the upper left of *Fig.* 38.

The charts found in *Figs.* 37 and 38 make possible a quick estimate of overall gear proportions. Using the practically established ratio of face width to pinion diameter, which ranges from 1:1 to 3:1, and certain AGMA standards regarding center distance and face width relations, it is possible to arrive at a reasonably accurate determination of gear size for high-speed application.

OTHER DESIGN CONSIDERATIONS: To realize the most effective design from the standpoints of efficiency, wear, etc., a careful study should be made of the limitations on materials, gear and pinion hardness, tooth runout, spacing, torsional and transverse deflections, etc., which are covered in AGMA Standard Practice.

Sound Level and Tooth Profile Accuracy: Regardless of the amount of thought devoted to careful gear design, the entire effectiveness of the drive can be destroyed by lack of manufacturing accuracy. Common specifications placed on high-speed gearing by users of this equipment are concerned with the sound level, *Fig.* 39, and the magnitude of vibration. Many high-speed gears, such as those driving centrifugal refrigerant compressors for air-conditioning service, *Fig.* 40, are used in office building installations, theaters, etc., where sound level is a major consideration in view of human comfort and efficiency. For the majority of such applications, operating at speeds of 5000 to 9000 revolutions per minute, a sound level of 85 decibels is considered satisfactory.

Probably the most important single factor influencing the sound level of a high-speed drive is the accuracy of the tooth profile. The maximum permissible error in tooth dimensions measured on the pitch circle is a function of velocity and is plotted in *Fig.* 41. From the chart a justified conclusion would be that an accuracy of 0.0005-inch should be sufficient for any speed in excess of 5000 feet per minute. As far as satisfactory operation with respect to quietness and load carrying capacity is concerned, such an accuracy is sufficient for a range of pitch-line velocities from 5000 to 10,000 feet per minute, except at critical speeds.*

The actual maximum error in gear tooth profile is a function of the class of gear and diametral pitch, *Fig.* 42. Study of this chart indicates that only

Nomenclature

C = Center distance, inches
d = Pinion diameter, inches
F = Face width of gears, inches
K = Service classification factor
M = R ÷ (R + 1)³ 31,500
n = Pinion speed, revolutions per minute
P = Horsepower transmitted per inch of face width
R = Gear ratio (number of teeth in gear ÷ number of teeth in pinion)
V = Pitch-line velocity, feet per minute
W = Tangential load transmitted per inch of face width

* Earle Buckingham—*Analytical Mechanics of Gears*, McGraw-Hill Book Co., New York, 1949, Page 388.

Fig. 39—Variation of sound level with speed. Intensity of sound is doubled with each 3-decibel increase

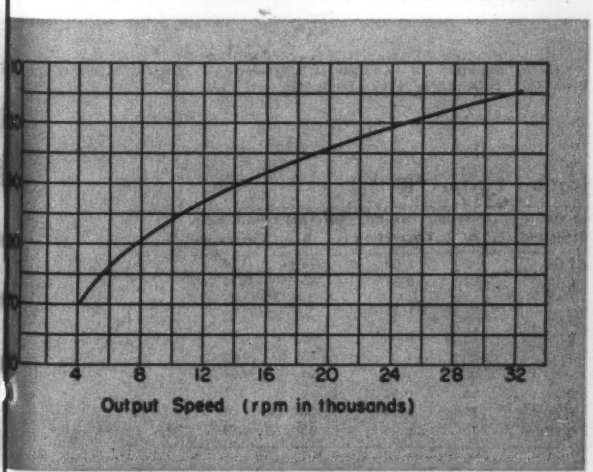
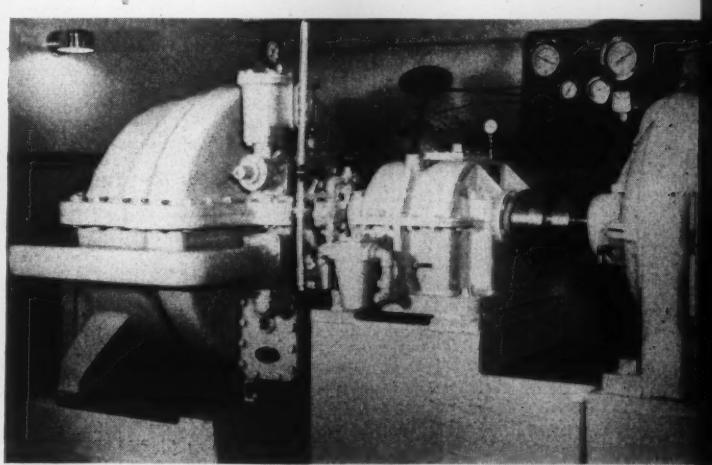


Fig. 40—A 350 hp, 6000 rpm gear driven centrifugal refrigerant compressor for air conditioning service



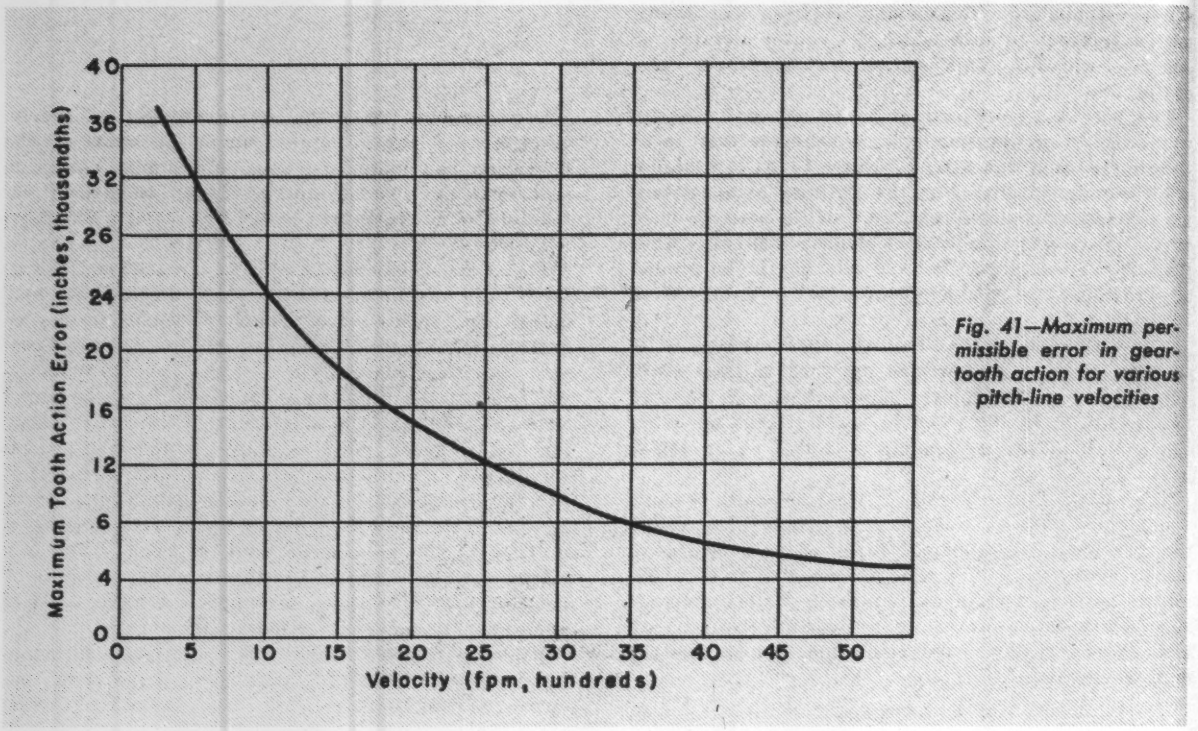


Fig. 41—Maximum permissible error in gear-tooth action for various pitch-line velocities

Class 3, precision-cut gears, would be satisfactory for high-speed operation. Another conclusion that can be drawn from the chart is that fine diametral pitches are necessary for high-speed operation. Present practice supports this conclusion†; finest pitches and steepest helix angles practicable are being used. Nearly all high-speed gearing is of the double helical design where there is no resultant thrust on the bearings and gear housing. By using steep helix angles the tooth overlap, or number of teeth in contact across the face of the gears may be increased. Where, in low-speed operation, helix angles of approximately 20 degrees for single helical gearing with tooth overlaps of 1¾ have been employed, in the high-speed range, helix angles as high as 45 degrees and tooth overlaps of 10 to 15 and above are being used. As a result, high-speed drives have many more teeth in contact simultaneously which means a more gradual transition of load and consequently a reduced noise and vibration level.

In order to lessen the degree of accuracy required, some attempts have been made to modify the tooth form. Such modifications should be kept to a minimum and should be made in the nature of a tolerance. In other words, the direction of permissible errors on tooth profiles should be in the direction that avoids edge contact at the beginning of mesh.*

It should be noted that there are other factors which contribute to the noise level. Gear blank form and material have an influence on the quietness of the gear. If the blank has bell-like characteristics it will pick up and sustain a sound of its own frequency that might be created by the gear teeth or by some outside action.

The design of the gear housing may also influence the noise of gears in operation. Some gear housings

† "High-Speed Industrial Gear Drives"—C. B. Connell, Westinghouse Electric Corp. Presented at the AGMA semiannual meeting, Oct., 1948.

* Earle Buckingham—*Analytical Mechanics of Gears*, McGraw-Hill Book Co., New York, 1949, Page 388.

Table 7—Service Classification Factor "K"

Driven Machine	Electric Motor K	Electric Motor Correction	Steam Turbine K	Steam Turbine Correction	Internal Comb. Eng. K	Internal Comb. Eng. Correction
Compressor (Centrifugal)	85	1.09	85	1.09	63	0.81
Compressor (Rotary)	63	0.81	63	0.81	55	0.71
Compressor (Refrigerant, air conditioning)	90	1.16	..	...	..	...
Fan (Forced draft)	85	1.09	85	1.09	70	
Fan (Induced draft)	75	0.96	75	0.96	63	0.81
Generator	..	...	110	1.41	63	0.81
Pump (Centrifugal, circulating)	95	1.22	95	1.22	75	0.96
Pump (Descaling, with surge tank)	55	0.71	55	0.71	..	...
Pump (Pipeline, centrifugal)	75	0.96	75	0.96	63	0.81
Pump (Waterworks, general purpose)	75	0.96	75	0.96	63	0.81

give resonance to sound while others do not. Backlash, if sufficient, may be the cause of excessive noise, heat and wear. Inadequate lubrication may also be a source of noise.

Notwithstanding, the most important factor is tooth profile accuracy. There are sufficient rewards for precise generation, provision for checking, and accurate checking machines, *Fig.* 43. This is borne out by the Westinghouse Laboratories who have taken these precautions and have installed and operated successfully, gears with pitch-line velocities as high as 23,000 feet per minute.

Lubricating and Cooling System: High speeds require, generally, the use of force-feed lubricating systems. Simple splash lubrication which is effective in industrial type gear reducers has not proved too effective in high-speed applications. The fact that the losses of a gear dipping in oil vary approximately as the square of the peripheral speed, make splash systems impractical.

Oil Serves Dual Purpose

The oil in a high-speed drive serves a dual purpose. In addition to acting as a lubricant, it removes the heat from the gears and the bearings. It becomes evident therefore that an oil circulating pump and heat exchanger are important parts of the cooling system whose function is to maintain a safe operating oil temperature. Oftentimes, the cooler must provide sufficient capacity to compensate for the insufficient capacity of the gearcase for thermal radiation, because the gearcase is relatively small compared to the horsepower transmitted in most high-speed gear applications.

The importance of maintaining proper oil temperature is illustrated in *Fig.* 44. During the test period from which this curve was drawn, the oil quantity was kept constant and the temperature controlled. Generally known is the fact that power losses increase as the oil viscosity increases. Consequently, it is to be expected that as the temperature is increased with a resultant decrease in viscosity, the power losses decrease. The curve bears this out. In the range of temperatures between 80 and 160 F, a 7½ per cent decrease in power requirement resulted. At the higher speeds, it should be noted, the power saving is greater. Viscosity effect is illustrated by the fact that the viscosity changed from 119 to 14 centipoises within the temperature range previously mentioned.

Power requirements vary more drastically with respect to the quantity of oil supplied to the gears, *Fig.* 45. Increased oil quantity results in a larger volume of the oil being sheared by the journal, and a greater quantity of the oil being squeezed in the mesh between the gear teeth. It is to be expected therefore that appreciable power demand would result from increasing oil quantity. The curve illustrates that the higher the speed, the steeper the curve

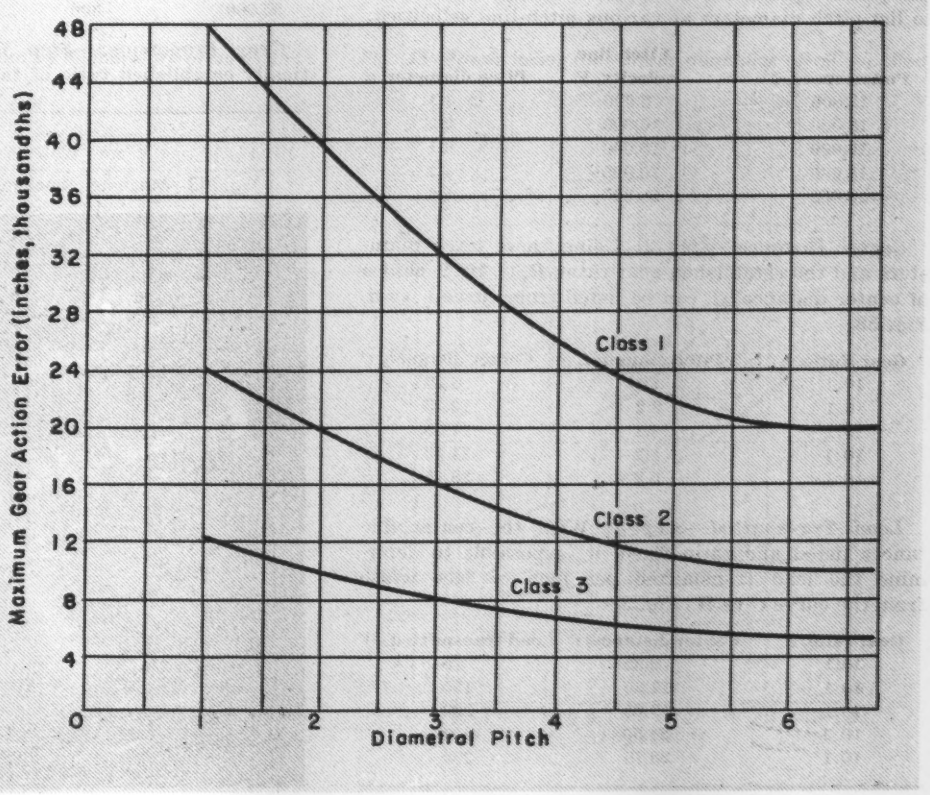


Fig. 42 — Actual maximum error permissible in gear action as a function of Class of gear accuracy and the diametral pitch

slope for any given quantity of oil. The power loss variation is almost as the square of the oil supplied as the higher rates of flow are reached.

Conclusion drawn from the study of *Figs.* 44 and 45 would be that not only should the quantity of oil be kept to a reasonable minimum, but that it would be better to use a small quantity of comparatively cool oil rather than large quantity of high-temperature oil. Actual testing of specific drives is the only method of arriving at the proper quantity and temperature. An interesting curve related to this determination is shown in *Fig.* 46. The optimum quantity is easily ascertained as 2¼ gallons per minute.

(*Continued on Page* 176)

Gear Calculations With Charts

USE OF CHARTS IN SELECTING HIGH-SPEED GEARING: To illustrate the use of the charts, two problems relative to high-speed drives are here presented. Basic data are as follows:

1. Prime mover—Synchronous motor, 1800 rpm, 200 hp
2. Driven mechanism—Centrifugal compressor, 18,000 rpm
3. Drive specifications—R = 10:1, K = 85 (TABLE 7).

First Method

Pitch Diameter—Step 1: With an rpm of 18,000 established as the compressor speed, and consequently the pinion speed, the d vs n curve, *Fig.* 37, is used to list pitch diameters at various pitch-line velocities:

Pinion speed, c	Pitch-line velocity, V	Pitch diameter, d
18,000	5,000	1.1
18,000	10,000	2.2
18,000	15,000	3.1
18,000	20,000	4.2
18,000	25,000	5.2

Center Distance—Step 2: Using these pitch diameters and the established gear ratio, R, of 10:1, values of center distance, C, can be listed from curve C vs d, *Fig.* 38:

Gear Ratio, R	Pitch diameter, d	Center distance, C
10:1	1.1	6.25
10:1	2.2	12.25
10:1	3.1	17.25
10:1	4.2	24.50
10:1	5.2	28.75

Load Transmitted—Step 3: With the center distances listed and ratio fixed, it is possible to determine the load transmitted per inch of face width from the curve C vs W, *Fig.* 38:

Gear ratio, R	Center distance, C	Load transmitted, W
10:1	6.25	75
10:1	12.25	155
10:1	17.25	220
10:1	24.50	320
10:1	28.75	370

Correction Factor—Step 4: Service classification factor, K, is greater than 78 and therefore TABLE 7 must be consulted for the correction factor:

Correction Factor	Load Transmitted, W	Corrected Load Transmitted, W
1.09	75	82
1.09	155	169
1.09	220	240
1.09	320	349
1.09	370	403

Horsepower—Step 5: By using the P/F vs W curve, *Fig.* 38, the load transmitted per inch of face width can be converted into horsepower transmitted per inch of face width:

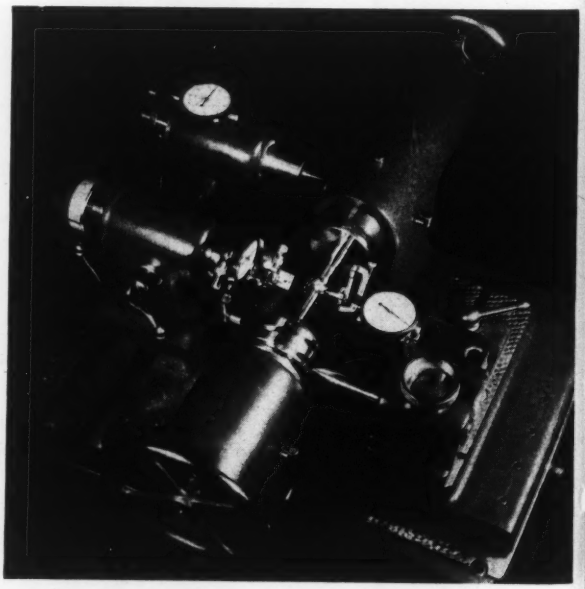
Pitch-line Velocity, V	Load Transmitted, W	Horsepower Transmitted, P/F
5000	82	12.5
10,000	169	50.0
15,000	240	113
20,000	349	213
25,000	403	300

Face Width—Step 6: From the data originally available it is known that 200 horsepower are to be transmitted. Knowing the capacity of the gears per inch of face width, it is possible to establish a list of face widths

Pitch-line velocity, V	Horsepower transmitted, P/F	Horsepower (total)	Face width, F
5000	12.5	200	16
10,000	50.0	200	4
15,000	113	200	1.77
20,000	213	200	0.94
25,000	300	200	0.67

Gear Proportions—Step 7: Employing the practically established ratio of face width to pinion diam-

Fig. 43—Machine checking gear form accuracy



eter, 1:1 to 3:1, it is possible to select the gear proportions most satisfactory of those listed:

Pitch-line velocity, V	Pitch diameter, d	Face width, F	Ratio, F/d
5000	1.1	16	14.6
10,000	2.2	4	1.82
15,000	3.1	1.77	0.57
20,000	4.2	0.94	0.224
25,000	5.2	0.67	0.128

Conclusions: Only the gear operating with a pitch diameter of 2.2 falls within the ratio limits specified. It might be concluded therefore that the gearing generally satisfactory will have the following approximate over-all proportions:

1. Center distance ..... 12.25 inches
2. Pitch diameter, pinion ..... 2.2 inches
3. Pitch diameter, gear ..... 22 inches
4. Face width, gears ..... 4.0 inches

Second Method

For the sake of presenting another method of attack including the charts, it can be assumed that because of certain space limitations, the center distance of the gears is restricted to 12.25 inches. Using the original data of the foregoing problem, the procedure is as follows:

Pitch Diameter—Step 1: With the ratio fixed at 10:1 and the center distance at 12.25 inches, it is possible by using curve C vs d to establish the pitch diameter of the pinion, $d = 2.2$ inches.

Load Transmitted—Step 2: With c and R fixed it is also possible to immediately establish the load transmitted per inch of wace width, $W = 155$ lb/in.

Correction Factor—Step 3: As before this load must be corrected by using the factor listed in TABLE 7 for a K value of 85 or, $1.09 \times 155 = 169$ lb/in.

Pitch-Line Velocity—Step 4: In order to convert load transmitted to horsepower transmitted, pitch-line velocity must be determined. Knowing the pinion speed to be 18,000 revolutions per minute, from the original data, and knowing from Step 1 that the pinion pitch diameter is 2.2 inches, it is possible to establish the pitch-line velocity by consulting *Fig.* 37. Thus, $V = 10{,}000$ fpm.

Horsepower Transmitted—Step 5: The horsepower transmitted per inch of face width at 10,000 fpm and load transmitted of 169 lb/in. is found on the P/F curve, *Fig.* 38 to be $P/F = 50$ hp/in.

Face Width—Step 6: In order to transmit 200 horsepower a face width of 4 inches necessary, $200/F = 50$ or $F = 4$.

Face Width Ratio—Step 7: As a check to determine the suitability of the gear, the ratio of face width to pinion diameter is calculated: $F/d = 4.0/2.2 = 1.82$.

Conclusions: If this ratio falls between the limits 1:1 to 3:1, specified as representing usable combinations, then the gear set has met the general requirements for high-speed application.

Fig. 44—Power loss as affected by temperature of oil to unit

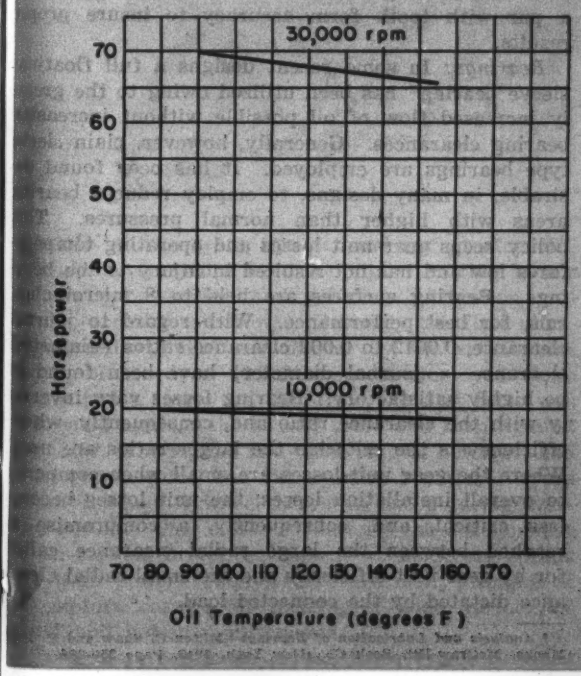
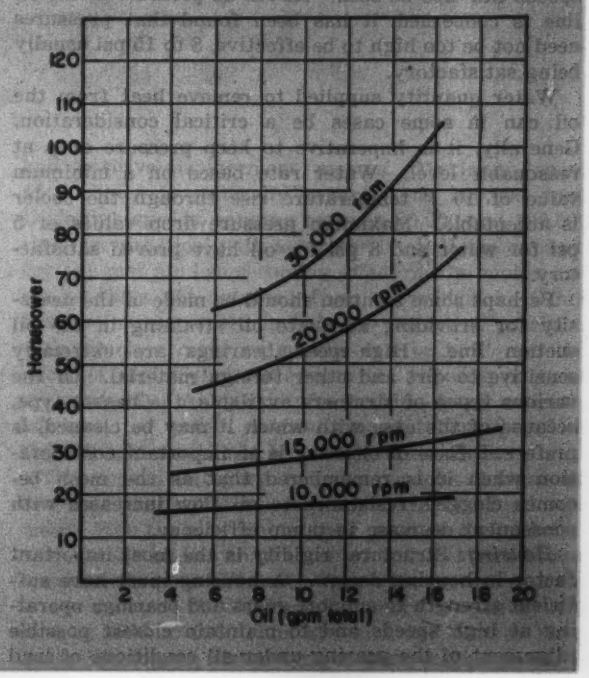


Fig. 45—Power losses for various quantities of oil supplied



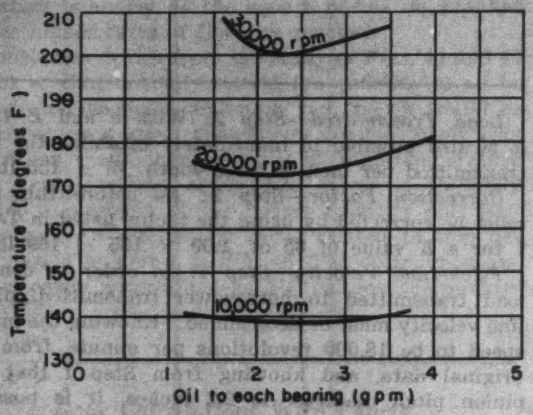


Fig. 46—Above — High-speed bearing temperatures for various quantities of oil supplied

Fig. 47—Right—Variation of losses with speed for high-speed drive, stressing negligibility of loading and prominence of speed with regard to losses

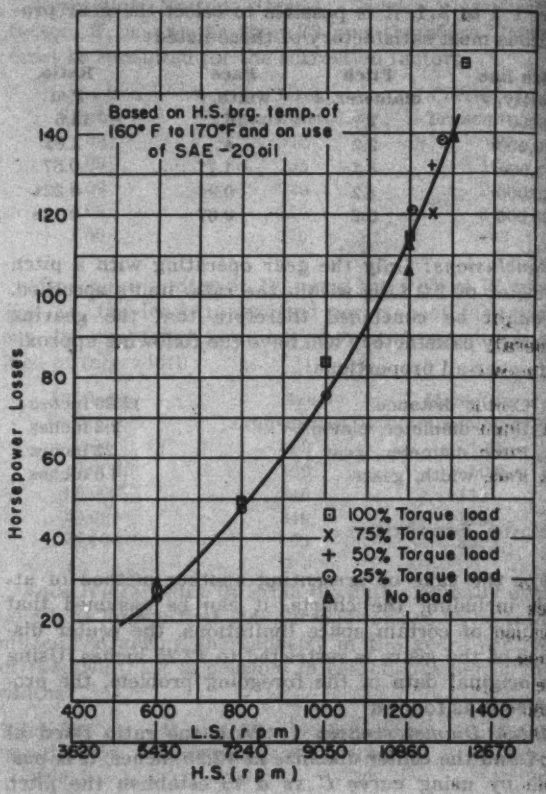


Fig. 48—Extreme Right—Efficiency-speed curve for high speed drive. It can be seen that loading has comparatively negligible effect on efficiency

Either side of this point shows increasing bearing operating temperatures. From the chart the relative increase of bearing temperatures with increasing speeds can also be seen. As far as pressure in the oil line is concerned, it has been found that pressures need not be too high to be effective, 8 to 15 psi usually being satisfactory.

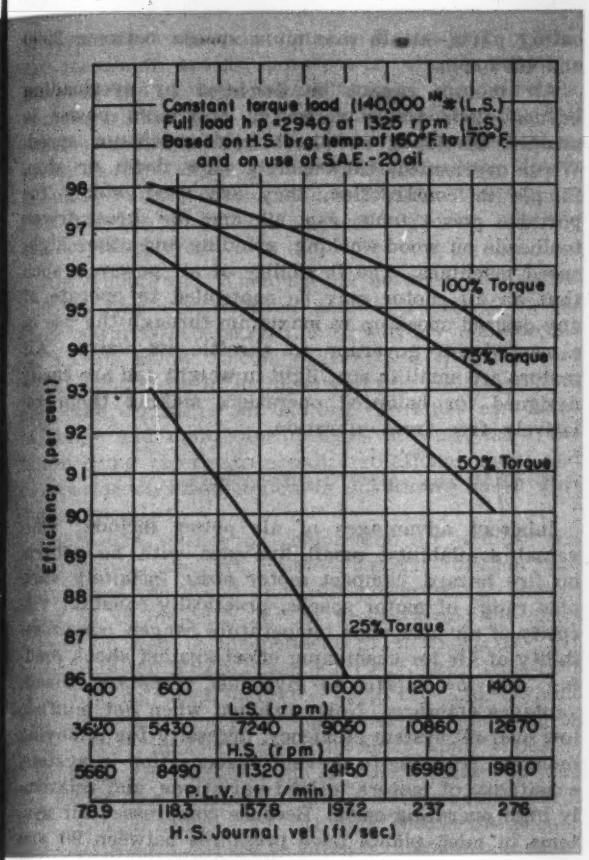
Water quantity supplied to remove heat from the oil can in some cases be a critical consideration. Generally, it is imperative to keep pressure drop at reasonable level. Water rate based on a minimum value of 10 F temperature rise through the cooler is acceptable. Maximum pressure drop values of 5 psi for water and 8 psi for oil have proved satisfactory.

Perhaps some mention should be made of the necessity for providing adequate oil straining in the oil suction line. High-speed bearings are extremely sensitive to dirt and other foreign material. Of the various types of strainers available, the basket type, because of the ease with which it may be cleaned, is preferred. Ease of cleaning is an important consideration when it is remembered that as the mesh becomes clogged, resistance to oil flow increases with consequent decrease in pump efficiency.

Housing: Structural rigidity is the most important factor in housing design. A housing must have sufficient strength to support gears and bearings operating at high speeds and to maintain closest possible alignment of the gearing under all conditions of load application. Materials often used are close-grained gray cast iron or fabricated steel, amply reinforced. The housing, in order to provide parallelism of shafts, must be carefully machined. Accuracy should be on a par with tooth form accuracy to insure proper results.

Bearings: In some recent designs a full floating-sleeve bearing* has been utilized owing to the greatly increased flow of oil possible without increasing bearing clearances. Generally, however, plain sleeve type bearings are employed. It has been found desirable, in many designs, to employ reduced bearing areas with higher than normal pressures. This policy keeps gear unit losses and operating temperatures low and has not resulted in injury to the bearings. Bearing surfaces are held to 8 microinches, rms, for best performance. With regard to journal clearance, 0.0012 to 0.002 clearance ratios (diametral clearance ÷ journal diameter) have been found to be highly satisfactory. Bearing losses vary inversely with the clearance ratio and, consequently, where efficiency is the criterion the larger ratios are used. Where the gear unit losses are small when compared to overall installation losses, the unit losses become less critical, and, consequently, a compromise is reached between the large radial clearance called for by gear unit efficiency and the small radial clearance dictated by the connected load.

* *Analysis and Lubrication of Bearings*—Milton C. Shaw and E. Fred Macks, McGraw-Hill Book Co., New York, 1949, Page 350-388.



Perhaps it should be emphasized that great care be exercised in the design of sleeve bearings for high-speed applications. Such bearings are a great deal more sensitive to changes in clearance and viscosity, to foreign materials in the lubricant, to misalignment, and to design or operating variations than are the bearings that operate at moderate speeds.

Seals and Breathers: Prevention of oil leakage from the gear drive requires suitable seals and drain-back holes. Oil leaving the journals as end leakage must be returned to the gearcase sump immediately. Escape of oil vapor from the gear case is prevented by use of a large breather which permits maintaining of low pressure inside the gearcase. Drainback holes in the breather provide for the return of the condensed oil to the gear sump.

GEAR UNIT EFFICIENCIES AND APPLICATION: As a unit operating at high speeds, the gear drive cannot claim the usual gear efficiencies of 98 or 99 per cent. This is borne out by *Figs.* 47 and 48. When it is considered that the majority of the losses in a high-speed gear drive are those of the bearings—which increase as the square of the speed—it becomes evident that efficiencies well below 98 per cent will have to be acceptable, at least until the bearing problem is solved. Nevertheless, high-speed gear drives have found wide application, and will continue to wherever required speeds in excess of prime mover operating speeds canont be obtained satisfactorily by any means offering comparable results.

Air Motors

SOME thirty years ago demand for greater efficiency created a need for higher cutting speeds in woodworking and grinding. Attention naturally turned to air power. Probably the first ultra-high-speed spindle was driven by an air turbine having a speed of 30,000 rpm. It employed three stages and a single jet. Air from the first rotor was redirected against the second rotor and then against the third. The second and third stages, however, had little power value but did equalize rotational speed by serving as a flywheel to sustain speed during overload periods. This motor encouraged the development of air-turbine driven equipment at moderate speeds and high power. Today, there are many designs of air motors capable of meeting a wide variety of speed and power requirements.

Air motors are made in two basic types, turbines and positive-displacement motors. Maximum speed of turbines is generally regarded as 100,000 rpm. Ultrahigh speeds, ranging between 30,000 and 100,000 rpm, are practical with a turbine because the expanding exhaust air has a cooling effect on the motor and its bearings not obtained from other drives. Other factors limiting speed include the degree of precision with which a rotor or spindle can be built and the effect of centrifugal force upon the rotating parts of the assembly. As speeds of a turbine drive increase, however, the output naturally must be small and at 100,000 rpm would not be over ¼-horsepower.

The second category includes various designs of vane and piston motors. Although designed for lower speeds than the air turbine, their starting and running torques are higher than those of turbines and motor operation is more efficient. Vane type motors operate at speeds up to 25,000 rpm, but more frequently are used at 3000 to 5000 rpm, at which speeds they have better efficiency. Piston type motors, having recipro-

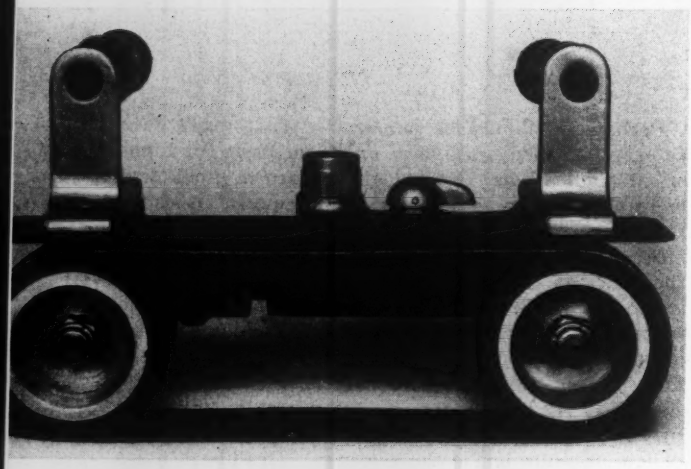
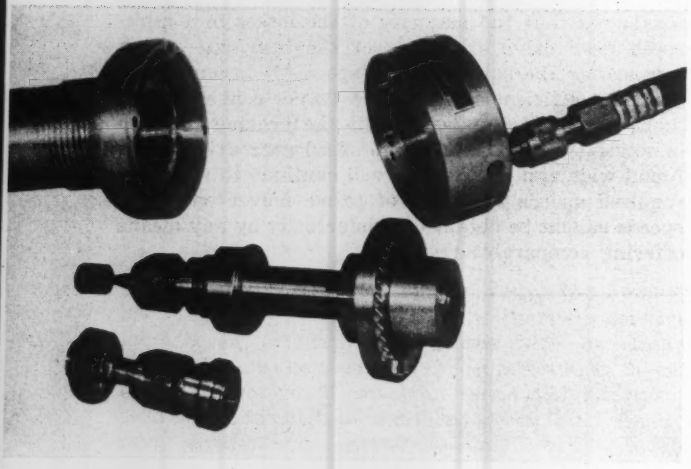
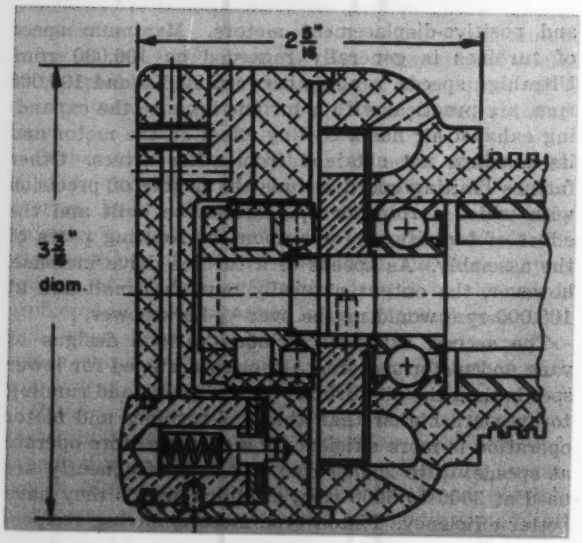


Fig. 49—Above—Belt sander having compact air motors in each pulley for the sanding belt

Fig. 50—Below—Disassembled view of turbine drive with a speed control governor for 50,000 rpm. The turbine is rated at one-half horsepower





cating parts, attain maximum speeds between 2000 and 4500 rpm.

Air motors cannot be damaged by overloading beyond their power capacity. Maximum power is usually developed at about half maximum speed. When overloaded, they simply slow down or stop. Simple in construction, they are used widely for portable power tools, *Fig.* 49, and for direct-driven toolheads on wood-working, grinding and other high-speed machines. The flexibility of air power is such that an air motor may be controlled to operate at any desired speed up to maximum through the use of a centrifugal governor to throttle its valve. Air motors are small in size, light in weight and are easily designed for balanced operation, making them relatively free from vibration.

Advantages and Limitations

Inherent advantages of air power include: Universal availability, small line size with no return, no fire hazard, compact motor sizes, infinitely variable range of motor speeds, practically constant viscosity of air over wide temperature ranges, compressibility of air for cushioning effect against shock loading, and low inertia to give fast response. Disadvantages embrace: Noisy exhaust when not muffled, low over-all system efficiency, necessity for removing moisture from air to prevent condensation in system, sensitivity of motors to load variations, and relatively high operating costs. Because compressed air systems in most plants have pressures between 90 and 125 psi, air motors are designed to operate within these working pressures and are usually rated at 90 psi. A motor, built to operate at 90 psi, however, may lose 25 per cent of its capacity if line losses in delivering the air to it drops the pressure to 70 psi.

Maximum power is developed at about one-half free speed. Increasing the line pressure to an air motor increases its maximum power and torque in proportion. Also, free speed and speed at maximum power are increased somewhat. In other words, if a motor is rated for 1 hp at 100 psi line pressure, it will develop about 2 hp at 200 psi.

With respect to light weight and small size, air motors have definite advantages. Comparing one horsepower motors used for portable tools, universal electric motors weigh about 12 pounds, 180-cycle drives weigh 7 pounds and 360-cycle units weigh 3 pounds, while air motors average 3 to 5 pounds.

Speed of air motors is infinitely adjustable by controlling the opening of the inlet air valve. This may be done manually, but for applications requiring speed control a centrifugal type of governor or a tachometer may be employed to adjust the valve. Speed may thus be held constant within the capacity of the motor and the sensitivity of the governor. Some motors have built-in governors to maintain rated speed and to prevent overspeed but not to regulate for reduced speed.

Characteristics of air motors, based on commercial

Fig. 51—Left—Sectional view of turbine unit developing one-half horsepower at 42,000 rpm

ly available designs, are listed in TABLE 8. Tabulated are maximum speeds, horsepower, air consumption, torques and general characteristics for turbines, rotating sliding-vane motors and piston motors. For continuous duty, five horsepower of compressor capacity is required for each motor horsepower. Motors generally are operated intermittently, however, so that the use of accumulators reduces the required system capacity.

AIR TURBINES: When speeds between 30,000 and 100,000 rpm are required and where speed rather than power is necessary, the air turbine is employed. At speeds below 15,000, other types of drives are more economical, including high-cycle electric motors and other types of air motors. Turbines are made in single and multistage designs, advantages of the latter being chiefly to equalize rotational speed and to provide flywheel action in sustaining speed during overload.

Controlled release of compressed air against the air turbine impellers creates spindle rotation and determines the speed and horsepower of the motor. Flow of air depends on the pressure and jet characteristics and is independent of rotor speed. Air turbine design makes possible a centralized drive that is easily balanced, operates smoothly and without vibration. Its simplicity, light weight and coolness in operation make it useful for many applications.

Singular advantage of the air turbine motor is that the expanding exhaust air has a refrigerating effect that keeps bearings cool. At high speeds the proper lubrication of bearings is often a difficult problem, but with turbines the cooling effect of the expanding air during exhaust makes bearing operation practical at 100,000 rpm without external cooling methods.

Turbines Sensitive to Load

Illustrated in *Fig.* 50 is a disassembled view of an air turbine. It has an automatic governor to maintain a speed of 50,000 rpm. This motor, about 7 inches in length, is one-third the size of the smallest ½-hp standard electric motor. In *Fig.* 51 is a sectional view of the same type of turbine, delivering ½-hp at 42,000 rpm. Because turbines are sensitive to load, an automatic governor maintains turbine speed under load. An expansion-ring type of governor, it controls the valve in a secondary jet.

When speed is below governor speed, both jets are open as indicated in the diagram in *Fig.* 52. This auxiliary jet either carries the load or aids in bringing the turbine up to governor speed. After governor speed is reached centrifugal force on the flexible element moves the governor sleeve to the left as shown in the diagram. This action cuts off the air supply through the bleeder line and by differential pressure closes the valve of the secondary air jet.

In *Fig.* 53 is a small 1/6-hp turbine developing 75,000 rpm. Maximum speed at which this motor operates depends upon its use and is controlled manually by valve adjustment. In *Fig.* 54 is a 75,000 rpm turbine. Its speed also is manually adjusted by the valve opening. This motor unit, designed for a pencil grinder, weighs 8 oz and develops an estimated 1/10-hp at 75,000 rpm and is designed in a hand tool for ⅜-inch diameter grinding wheels. In these turbines, full speed is obtained within two or three seconds. Without a braking device, normal deceleration by friction alone, depending on the speed at which the motor is operating, requires from 40 to 60 seconds.

Of particular interest is a turbine drive for a high-current disk generator, *Fig.* 55, in which the turbine buckets are an integral part of the disk rotor. The drive was developed by R. I. Strough, Case In-

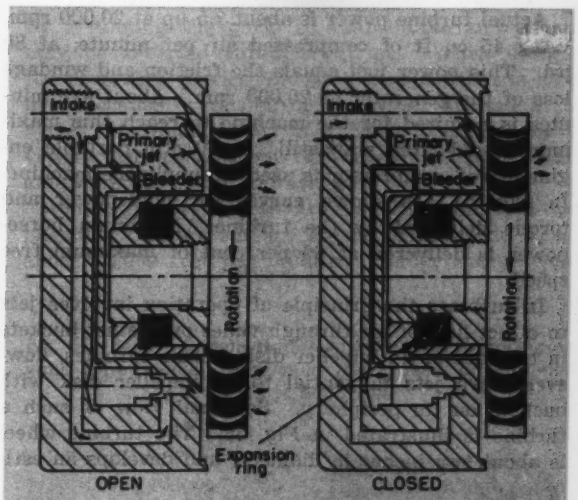
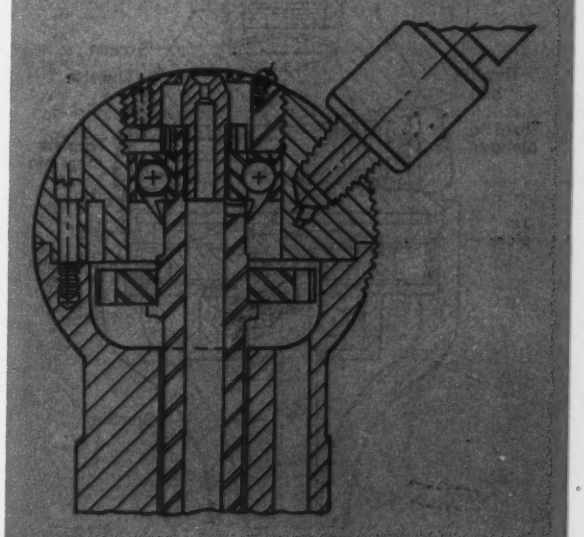


Fig. 52—Above—Expansion-ring governor controls an auxiliary jet to maintain speed under load. Heavy-load condition with secondary jet open is shown at left, light-load operation is at right

Fig. 53—Below—Section view of small turbine-driven tool rated at 1/6-horsepower and 75,000 rpm



stitute of Technology. Its actual efficiency is of no importance in studying the principles of the generator which is a direct-current machine of the series unipolar type having no iron in its magnetic circuit. It produces pulses of high magnetic field and is operated short circuited. The rotor, serving as a flywheel for energy storage, is driven up to speed between current pulses by a jet of compressed air acting on the turbine buckets cut into the upper surface of the rotor, *Fig.* 56.

The rotor is forged Berylco 25S alloy. The forging was rough turned and the turbine buckets milled in its upper surface,-*Fig.* 6, with a conventional end mill. The rotor was then heat treated, finish ground and balanced. The bucket recesses were not ground. The nozzle is machined in a 24 ST Dural block and is a Laval type having a throat area 0.025 square inches.

Actual turbine power is about 1.5 hp at 20,000 rpm, using 45 cu ft of compressed air per minute, at 80 psi. This power just equals the friction and windage loss of the generator at 20,000 rpm. About five minutes is required for the machine to reach this maximum speed from standstill. A 60 cfm portable engine-driven compressor is used to supply the machine. In *Fig.* 57 are shown curves for horsepower and torque delivered by the turbine. Maximum horsepower is delivered at 60 per cent of maximum free speed.

In turbines the principle of operation involves jets to direct flow of air through vanes or against buckets in the face of an impeller disk. Some turbines, however, have jets tangential to an impeller disk with buckets on its periphery. Sectional view of such a turbine is illustrated in *Fig.* 58. The turbine wheel is about two inches in diameter and develops an estimated ¼-horsepower at 50,000 rpm and 20 psi air supply. Air consumption is 26 cfm. Lubrication is introduced through the turbine head cap above the center of the spindle, flows over the bearing and is delivered into the turbine exhaust. Vanes below the turbine wheel create an upward draft in the flared insert, drawing air in and around the spindle and exhausting it through the turbine exhaust. This prevents oil or condensate leakage down the spindle into the driven element.

SLIDING-VANE MOTORS: At speeds under 25,000 rpm sliding-vane motors are utilized, particularly where heavier loads are encountered, when operation is in hazardous locations or where a compact and lightweight drive is essential. Principle of operation

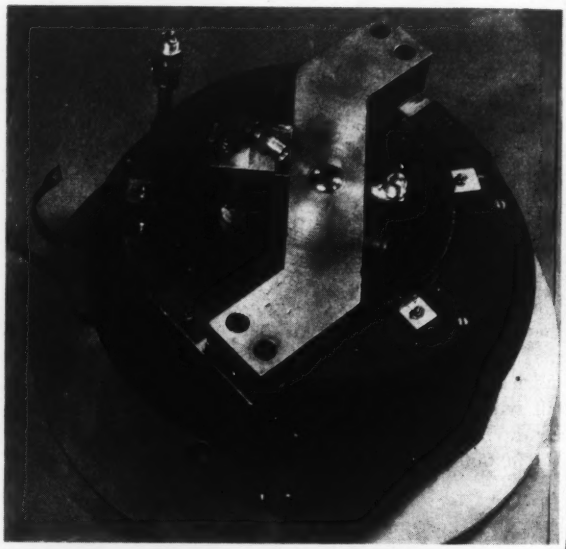


Fig. 54—Below—For a pencil grinder weighing only 8 oz, this turbine has a free speed of 75,000 rpm

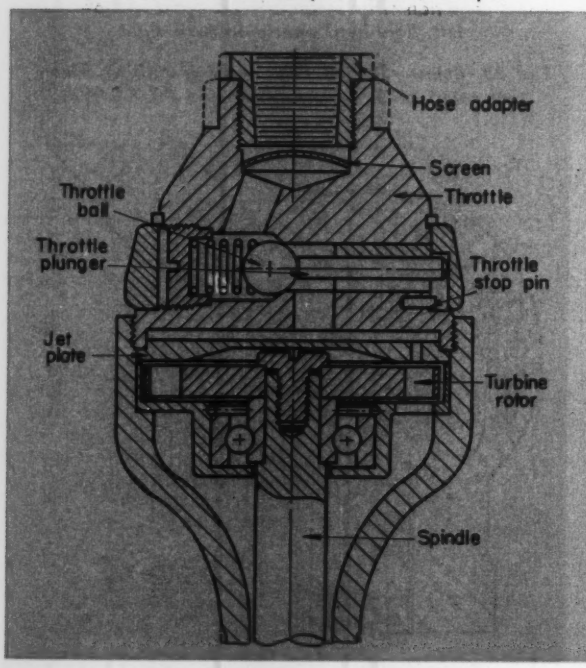
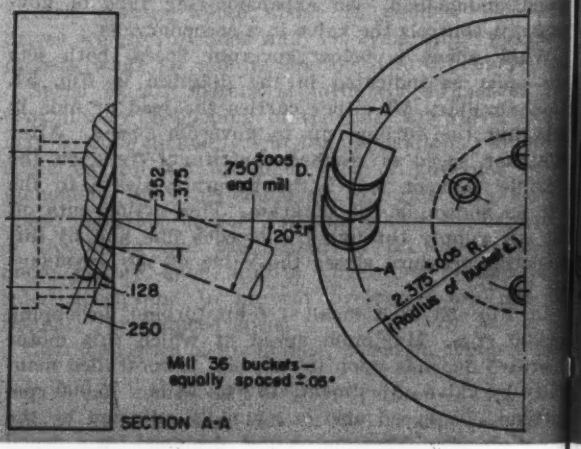


Fig. 55—Above—Novel experimental pulse generator has turbine buckets machined in the generator rotor, obviating additional rotating parts for the drive. Turbine is unharmed by sudden deceleration

Fig. 56—Below—Details of end-milled buckets in rotor of experimental generator



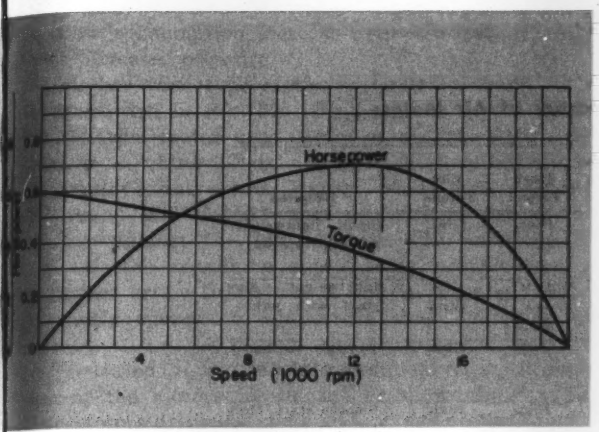


Fig. 57—Above—Horsepower and torque characteristics of the turbine illustrated in Fig. 55

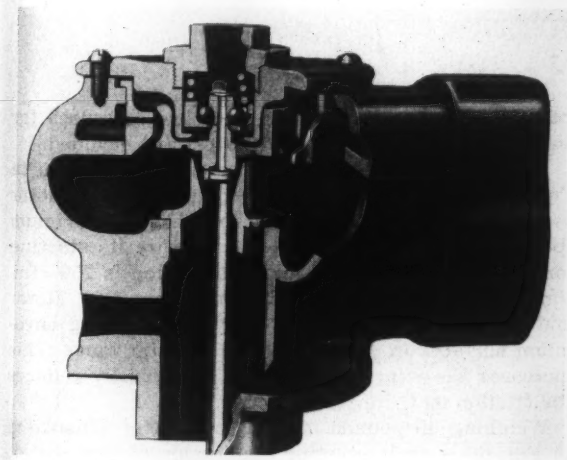


Fig. 58—Above—Turbine having tangential jets on the impeller. Unit drives a laboratory centrifuge at 50,000 rpm

Fig. 59—Below—Vane type motor sectioned transversely, showing sliding action of vanes in rotor and eccentric liner

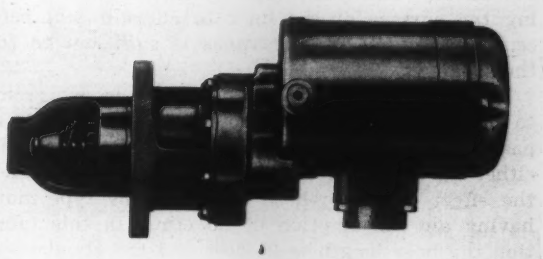
is shown in *Fig.* 59. Vanes, sliding in slots in a rotor form seals with an eccentric liner. Intake and exhaust ports are positioned so that the working stroke of each vane has about 3 to 1 expansion ratio.

Being a positive-displacement type, vane motors have more ability, compared with turbines, to carry loading without extreme changes in speed. Starting is not as smooth, however, in motors designed for the higher speeds. They have a tendency to break away with a jerk. Also, leakage in starting is high until the rotor has sufficient speed to seal the vanes against the liner through centrifugal force.

Vane motors may be designed for reversing duty by arranging the ports so that intake and exhaust may be reversed through valving. When designed for reversing service, economy of operation is sacrificed because expansion of air is not utilized in the cycle. Maximum speed of a motor designed for reversing service is about two-thirds the speed attainable were the motor nonreversing.

Vanes in air motors are usually laminated-duck phenolic, limiting surface speeds of the vane tips to about 10,000 feet per minute. Higher speeds would

Fig. 60—Below—Engine starter employing a 10-horsepower vane type motor. Maximum free speed of the motor itself is 6000 rpm



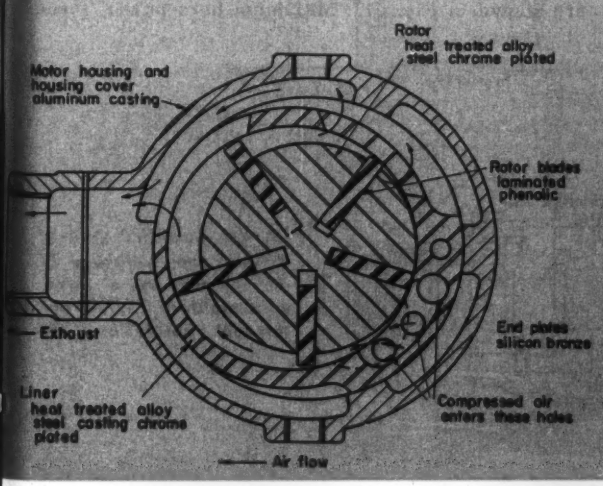


Fig. 61—Below—Typical characteristics of a sliding vane motor. The motor is rated 10 horsepower at 3000 rpm

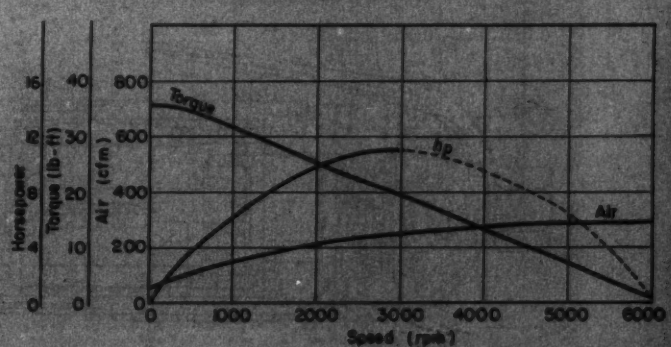


Table 8—Typical Characteristics of Air Motors

(at 90 psi)

Type Motor	Free Max. Speed (rpm)	Capacity Ranges (hp)	Air Consum.* (cfm/hp)	Average Weight (lb/hp)	Start Torque	Run Torque	Remarks
Turbine	100,000 to 22,000	0.05 to 3	200 to 30	14 to 3	Poor	Low	Small, lightweight for high speeds
Sliding Vane (rotary)	25,000 to 3000	0.15 to 20	100 to 20	20 to 3	Fair to Good	Good	Lightweight. May start with jerk
Piston (radial)	3000 to 1000	3 to 18	25 to 18	30 to 13	High	High	Difficult to balance. Pulsating drive
Piston (wobble plate)	4500 to 2000	0.5 to 2.5	21.5	10 to 7	High	High	Smooth start. More efficient at lower speeds
Piston (Berry type)	25,000	0.3 to 18	28 to 19	10 to 5	Good	High	No pulsations. High efficiency in larger sizes

* Free air compressed to 90 psi

cause charring and deterioration of the blades, resulting from friction between the blades and the motor liner. Three to five vanes are utilized in motor rotors, three vanes for the higher speeds. Better and more positive starting, however, is obtained with five vanes. To improve starting characteristics and reduce blow-by, some motors have passages drilled to bypass a small amount of air under pressure to the root of the vanes. This pushes the vanes outward, contacting the surface of the liner to effect a seal before centrifugal force on the vanes is sufficient to form the seal.

Adequate air supply to the motor is as essential to operation at capacity power as are porting and passages within the motor itself. Restrictions in either adversely affect performance. TABLE 9 shows the effect of supply-hose size for vane type motors having air consumption of 60 cfm. In this tabulation the hose length is 25 feet and the standard size of hose at 90 psi is 100 per cent to show the relative effect of smaller hose on the capacity of the motor. Relative losses only are shown, not overall losses. Also, the relative effect of low pressures for hose size on a motor rated at 90 psi are included.

A typical vane motor is shown in cross section in *Fig.* 59. Employing five vanes this particular motor develops 10 hp at 3000 rpm and has a maximum speed of 6000 rpm. At 100 psi pressure its starting torque is 35 pounds, and air consumption is 250 cfm. Vanes in this motor are laminated phenolic. Rotor and liner are chrome-plated alloy steel, having chromium surfaces in contact with the plastic vanes. This increases the wearing properties and greatly reduces the friction on the moving vanes.

Weighing 45 pounds, the engine starter illustrated in *Fig.* 60 is an interesting application of this motor. Employing a built-in reducer and Bendix drive, the starter is designed for cranking engines in hazardous locations and in extremely cold climates. No governor is employed, speed control being manual through valve position. A friction clutch protects the driven elements in the starter against damage. Motor performance curves for this starter at 100 psi are shown in *Fig.* 61. Maximum horsepower, torque,

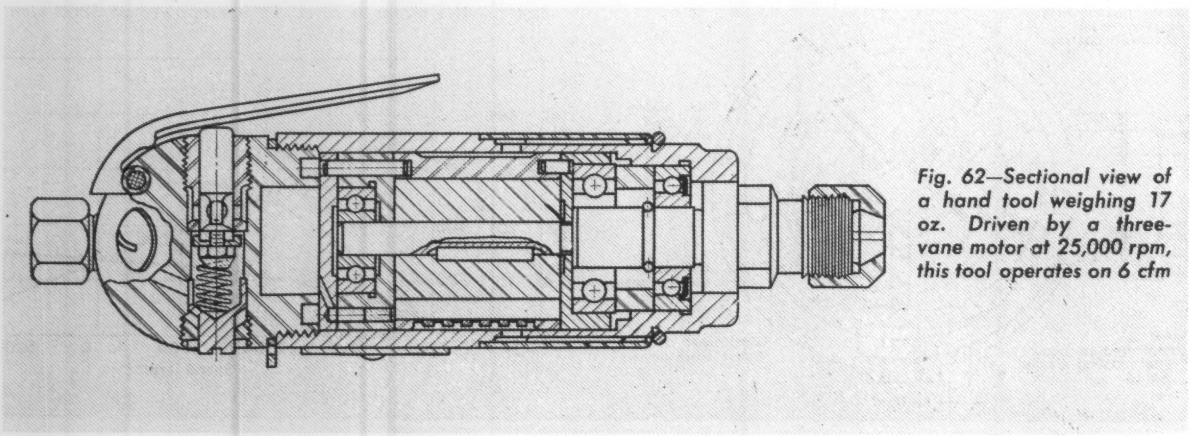


Fig. 62—Sectional view of a hand tool weighing 17 oz. Driven by a three-vane motor at 25,000 rpm, this tool operates on 6 cfm

and air consumption increase approximately directly with increased pressure.

Another vane type motor designed for hand tools is illustrated in *Fig.* 62. Designed for speeds up to 25,000, this motor uses 6 cfm of air, the entire assembly weighing 17 oz. Being a high-speed motor, it employs only three vanes, which are phenolic laminated and depend upon centrifugal action for sealing against the air pressure. No governor is utilized, the maximum speed being adjusted by a regulator screw to restrict the valve opening. Characteristic curves are similar in shape to those in *Fig.* 61.

Illustrated in *Fig.* 63 is a sectional view through one of the motors in the sander shown in *Fig.* 49. Two motors drive the portable tool, one on the spindle of each pulley, each developing ½-hp at 20,000 rpm. The speed of this three-vane motor is reduced to pulley speed of 6000 rpm through the planetary gearing shown. Air to the second motor passes through a telescoping tubular member and, with the aid of a compression spring serves to maintain belt tension between the two pulleys only during the periods when the belt is being utilized.

Governors to maintain rated speed and horsepower of a motor may be either of the expanding ring type shown in the turbine motor in *Fig.* 51 or of the flyweight type. One design of the latter type, for which a patent is pending is illustrated in *Fig.* 64. Assembled to the spindle of a vane motor, this governor has no parts which rotate with respect to each other. Speed at which the governor throttles the motor is

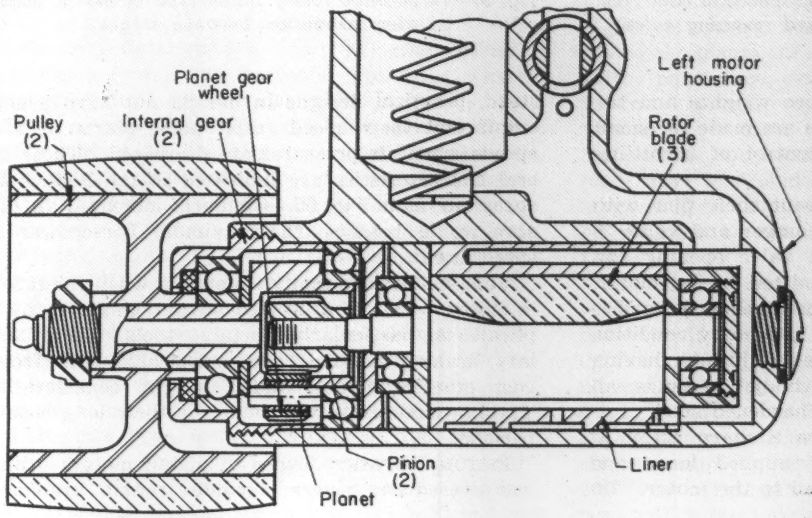
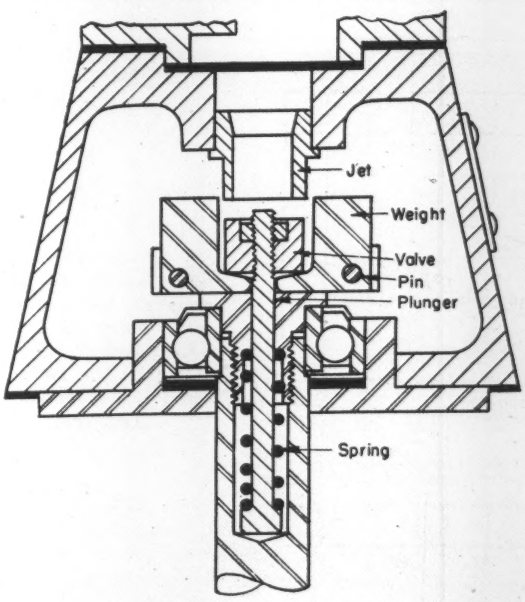
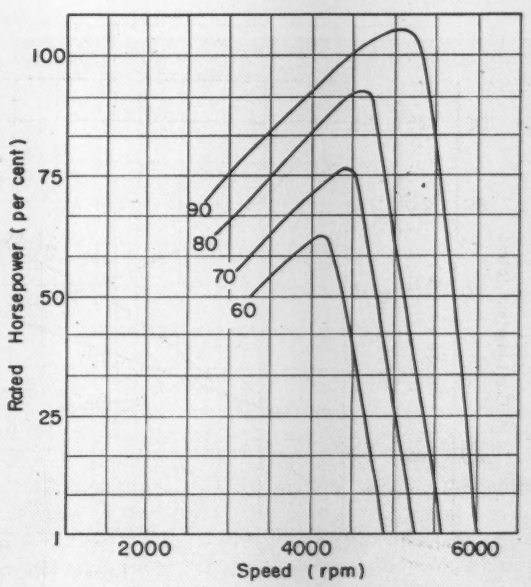


Fig. 63 — Left — Sectional view of a three-vane motor for driving the belt sander shown in Fig. 49. Planetary gearing reduces the speed of the motor

Fig. 64—Left, below — Flyweight governor designed to maintain speed and prevent overspeeding. Mounted on the motor spindle, parts have no relative rotary motion

Fig. 65—Below — Effect of air pressure on governor controlled vane type motor with governor set at 5000 rpm for 90 psi





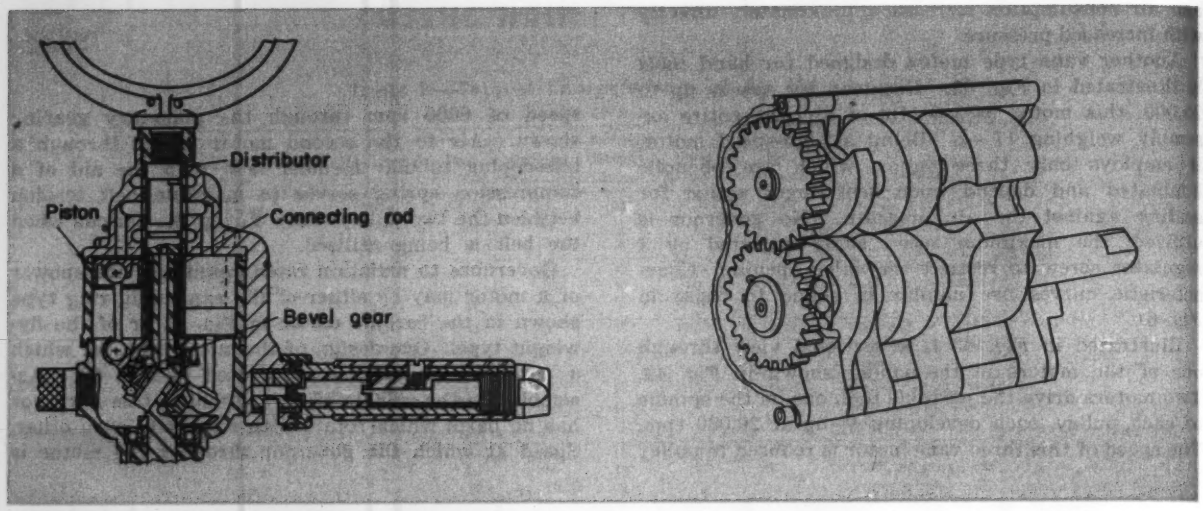


Fig. 66—Wobble-plate motors develop speeds to 4500 rpm. Design shown has four pistons and reversing valve

Fig. 67—Three-stage rotary piston type of motor, employing expansion between stages

determined by the size of the two weights and the spring on its plunger. Provisions are made for small adjustment of the spring for control of throttling action at desired speed.

The weights pivot outward about their pins with increase in speed, causing the plunger and valve to raise. This action restricts the valve opening and decreases the amount of air admitted to the motor. During throttling the valve never touches the orifice of the jet. If it were touching under any condition of operation, the motor would stop through having no air supply. In this design the jet is brass, all other parts of the governor are hardened steel.

An ideal governor would have a sharp cutoff at its speed setting regardless of applied loads and variations in air pressure supplied to the motor. Instead, practical designs in use do not have a sharp cutoff and their speed cutoff point starts at lower speeds when air pressures are decreased. These general characteristics are indicated in the horsepower-speed curves in *Fig.* 65. Similarly increased air pressure would increase the maximum horsepower and speed at cutoff.

Vane motors are positive stopping when the throttle is closed. This is a decided advantage for many applications, particularly portable tools, where auxiliary braking methods are undesirable both from a cost standpoint and added weight considerations. Turbine motors are notorious for their long coasting periods.

PISTON MOTORS: Types of piston motors utilizing compressed-air power include radial crankshaft,

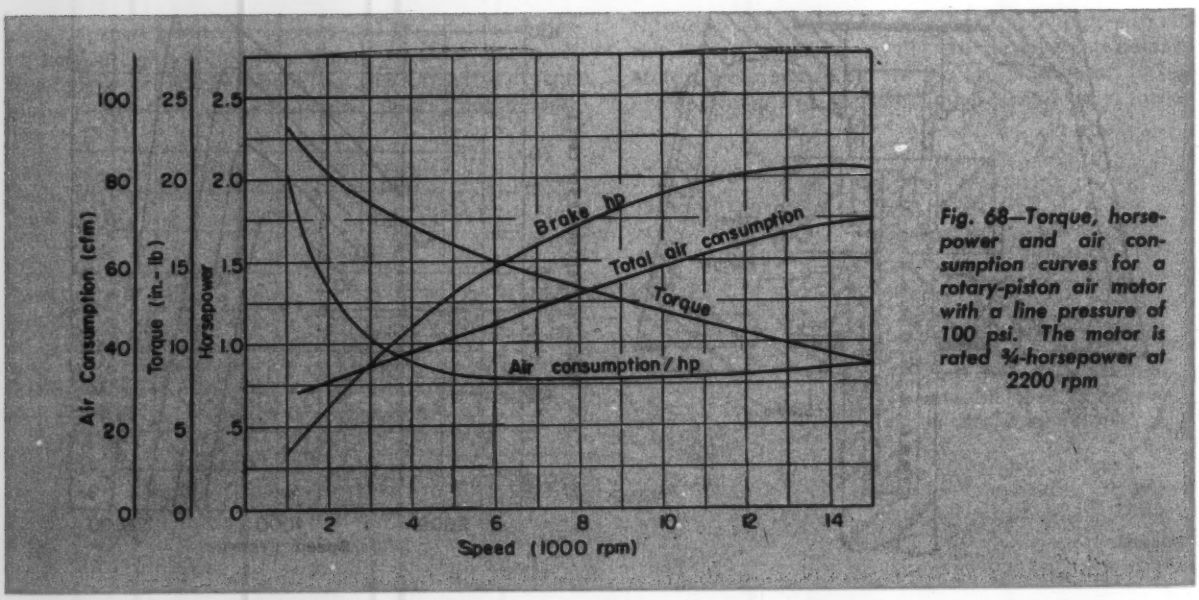


Fig. 68—Torque, horsepower and air consumption curves for a rotary-piston air motor with a line pressure of 100 psi. The motor is rated ¾-horsepower at 2200 rpm

Table 9—Effect of Hose Size on Power Developed

(Based on 25-ft length of hose and recommended size as 100% efficiency)

Air Consumption (cfm)	Hose ID (in.)	Fitting Size (in.)	Efficiency 90 psi (%)	80 psi (%)	70 psi (%)	60 psi (%)
15	⅜	¼	100	77	65	53
15	¼	¼	77	59	47	35
60	¾	½	100	90	75	61
60	½	⅜	86	62	51	39
60	⅜	¼	77	66	55	44

wobble plate and rotary piston designs. Air is not employed in conventional in-line crankshaft units, except possibly for slow-speed emergency service. Reciprocating piston motors of the radial crankshaft type are not considered suitable for high-speed applications. Although developing free speeds as high as 3000 rpm, they produce their maximum power and efficiency between 800 and 1000 rpm. Where horsepower requirements fluctuate rapidly or where close regulation of speed is desirable, governors control these motors within 25 rpm of the speed setting.

Piston motors develop higher torques at start and at slow speeds than do other types of air motors. Their pistons have better seals than do the vanes in vane motors. Cutoff can be advanced or retarded at will to vary the torque characteristics. Curves for horsepower characteristics vs speed are not symmetrical on each side of maximum power, like those shown for vane motors in *Fig.* 61. Instead the curves are steeper for speeds less than maximum horsepower than for speeds greater. Piston motors, like vane motors, may be designed for reversing service.

Wobble-plate type piston motors, *Fig.* 66, operate at speeds up to 4500 rpm. The motor in the hand tool illustrated has four 15/16-inch diameter pistons with a 15/16-inch stroke. The entire tool is 12 inches in length and weighs 5¼ pounds. Reciprocation of the pistons tilts the wobble plate progressively in the rotational direction of successive piston strokes. This progressive action of the wobble plate rotates the offset crank on the driveshaft and rotates the distributor or timing valve through a bevel gear on the wobble plate assembly. Compact design and arrangement of the cylinders close to the distributor make the intake passages short. This provides for increased speeds and efficiency by having a minimum volume in which the compressed air does no work.

Rotary-piston air motors are of an unusual design, the principle being indicated in *Fig.* 67. In this three-stage design, a piston resembling a single gear tooth on a cylinder rotates in contact with an abutment valve. This valve has a recess or bucket which mates with the piston and is synchronized with it by a pair of timing gears. Air enters the first stage, drives the piston through its power stroke and passes to the next stage, driving it and then the third stage in two expansion cycles. The design employs no valving other than that of the pistons themselves. To increase torque at low speeds, an adjustable bypass valve is provided between the inlet and second stages. This valve permits the flow of air under inlet pressure to the second stage as load conditions require.

Rated sizes range from 1/3 to 18 hp. Air consumption ranges from 28 cfm/hp for the smaller sizes down to 19 for the largest, making the motor an efficient design, particularly the large size. Simple in construction and pulsation free in operation, they attain speeds up to 25,000 rpm with 1/3 and ¾-hp sizes. Shown in *Fig.* 68 are torque, horsepower and air consumption curves for a motor rated ¾-hp at 2200 rpm. This motor develops over 2 hp at 15,000 rpm with a total air consumption of 70 cfm.

High Cycle Motors

WHERE high rotational speeds are found to be a necessity in design, use of high-cycle motors may offer the most practical solution. It must be said in all fairness, however, that these special motors are no panacea for solution of such problems. Many mechanical design difficulties attendant to their practical application must be properly dealt with to assure the desired results. High-cycle motors are a highly flexible and useful tool in achieving exceptionally high horsepower outputs per pound but require meticulous care on the part of the designer in putting this tool to work.

Cost, always a factor of tremendous importance in design, is high primarily owing to special equipment to provide the frequency desired. The important factor is, however, that where the necessary performance cannot be obtained by any other practical

means, the motor cost assumes little importance in overall machine cost, *Fig.* 69.

Actually, there is no great mystery surrounding the ability of high-cycle motors to develop high horsepower outputs with comparatively small amounts of active material. Horsepower may be expressed as the product of three factors: A constant, the speed of rotation, and torque. This relationship is as follows:

$$Hp = \frac{TS}{5252}$$

where the torque T, is in ft-lb and the speed, S, is in rpm. Thus, it is easily deduced that for a given torque, if the speed is raised, the horsepower developed is increased proportionately, *Fig.* 70.

This and the other characteristics of the polyphase induction motor, the most common power drive of industry, makes it ideally suited for high-speed applications, *Fig.* 71. The outstanding qualities may be outlined as follows:

1. Constant torque under full load
2. Nearly constant speed under full load
3. Speed variable or set according to the frequency and number of poles
4. Absence of all moving electrical contacts, brushes or commutators.

It therefore becomes practical to develop exceptionally high horsepower outputs by applying high frequency alternating potential to more or less conventional induction type motors. From the chart in *Fig.* 70 it can be seen that increase in speed and voltage as well as horsepower is proportional to the frequency employed, the torque remaining virtually constant. These specific characteristics, of course, apply to one motor type. It is especially important to keep this relationship in view wherever a wide variation in speed range is desired. For instance, a two-pole motor which develops 1 hp at 84,000 rpm would produce only ½ hp at 42,000 rpm.

For a specific purpose, a high-cycle motor can be designed to meet the extremes in design requirements. Since the ability to produce torque is largely a function of the weight of copper and iron utilized, it be-

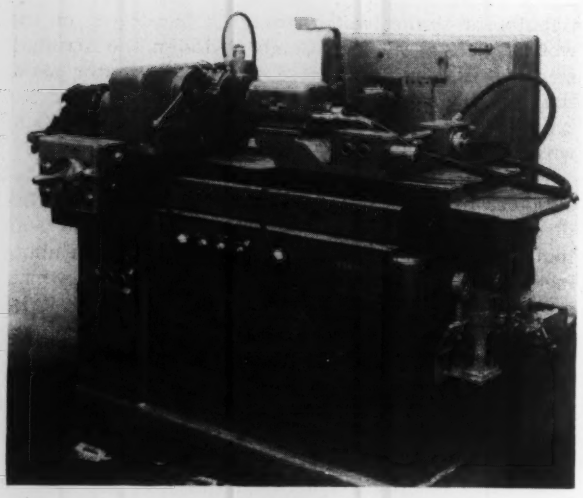


Fig. 69—Above—A Heald Machine Co. centerless grinder with a high-cycle wheelhead capable of speeds from 75,000 to 120,000 rpm on 1250 to 2000 cycles

Fig. 70—Below—Chart plotted to show the high-frequency characteristics of a standard 5 hp, 3600 rpm polyphase motor. Horsepower, voltage and speed vary directly with frequency, torque remaining constant. Amperage may decrease with increased frequency as efficiency is improved. Difference, however, is so small it is ignored

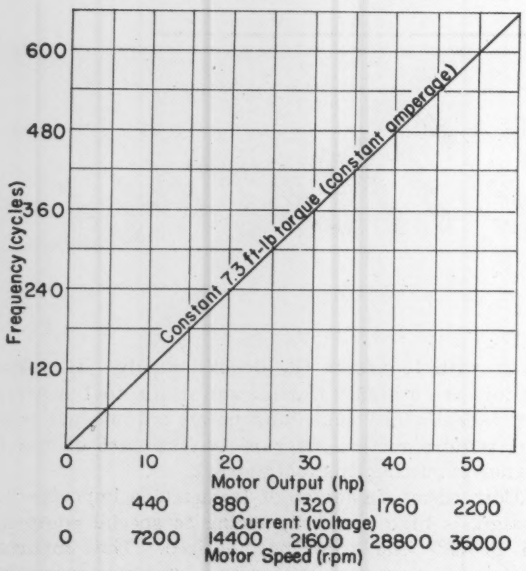


Table 10—NEMA Standard Voltages and Frequencies for Converters and Motors

Volts*	Frequency	Frequency Range
	Portable hand tools (8% regulation)	
110	30	25–60
110	60	60–125
110	90	120–180
110	120	180–240
110	180	180–360
110	360	360
	General purpose (15% regulation)	
110	30	25–120
110	60	60–240
110	120	125–480
110	180	250–720
110	360	400–1440
110	720	720–2880

* Attention is called to the fact that ratings have been selected so that the nominal voltage will be no higher than 220 volts for hand-tool applications and 440 volts for general-purpose applications.

Although the base voltage is shown as 110 volts, it is expected that the general practice will continue to be 220 volts for hand tool and 440 volts for general purpose, selecting the combination of frequency and speed from the list that will provide the desired rating.

At frequencies other than those listed above, voltage will be in direct proportion to the frequency.

comes possible by means of proper frequency and motor length to obtain a tremendous range of power output in a minimum of space, *Fig.* 72. The grinder motor shown in *Fig.* 73 indicates another extreme in the space factor suitability.

The no-load synchronous speed of an induction motor is equal to 120 times the frequency divided by the number of poles. Thus, the top speed of any given induction motor is a direct function of the applied frequency. Or

$$S = \frac{120F}{P}$$

where speed, S, in rpm is found from the frequency, F, in cycles per second and the number of poles, P. It is therefore possible to obtain direct-drive speeds by this method to the maximum practical and necessary in machines, *Fig.* 74. Frequencies to 2000 cycles per second and greater have been employed in practical applications, *Figs.* 69 and 75. Experimentally, speeds as high as 200,000 rpm have been attained.

Maximum speed of conventional 2-pole induction motors on standard 60-cycle frequency thus is 3600 rpm. Where this speed can be utilized in direct drive applications, *Fig.* 76, the additional and maintenance expense of special frequency can be avoided. However, where higher speed is imperative and belted drive is undesirable, special high-cycle motors provide the practical solution in a great many cases, *Fig.* 77.

In lieu of the more or less conventional design induction motor having constant-torque electrical characteristics, it is sometimes expeditious to utilize multispeed motor design. In contrast to the conventional induction motor design, multispeed motors employ special design to obtain two, three or four different constant speeds. For instance, two speeds may be obtained by means of two separate stator windings or by regrouping a single stator winding to give a different number of poles. With the former, any two possible speeds may be had, but with the latter the ratio is always 2 to 1. Three-speed motors have one winding reconnected to provide two speeds and a separate winding for the third speed. Four speeds are obtained by reconnecting both windings.

Unlike conventional induction motors, multispeed motors can be wound for constant torque, constant horsepower or variable horsepower and torque:

1. Constant Torque—Hp varies directly with speed
2. Constant Hp—Torque inversely proportional to speed
3. Variable Hp, Variable Torque—Hp and torque vary with speed, torque directly and hp as square of speed.

Any of these characteristics may be obtained by regrouping poles or by separate windings. Characteristic number 3 is seldom, if ever, required, the con-

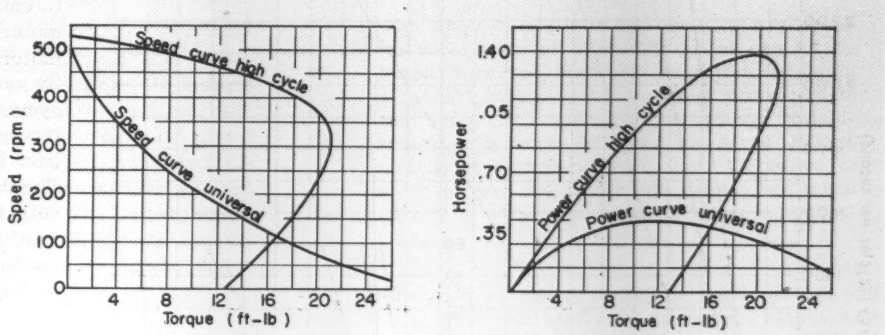


Fig. 71—Plots of speed curves and power curves of identically rated universal and high-cycle Black & Decker hand tools show comparative performance. Weight of high-cycle tool is much less than that of the comparable uinversal tool

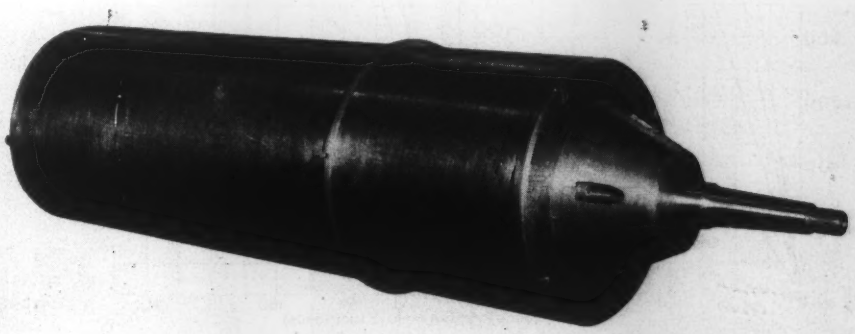


Fig. 72 — Wind tunnel motor of 250 hp output at 10,800 rpm. This compact 3-phase, 540-volt, 180-cycle induction motor is water cooled design

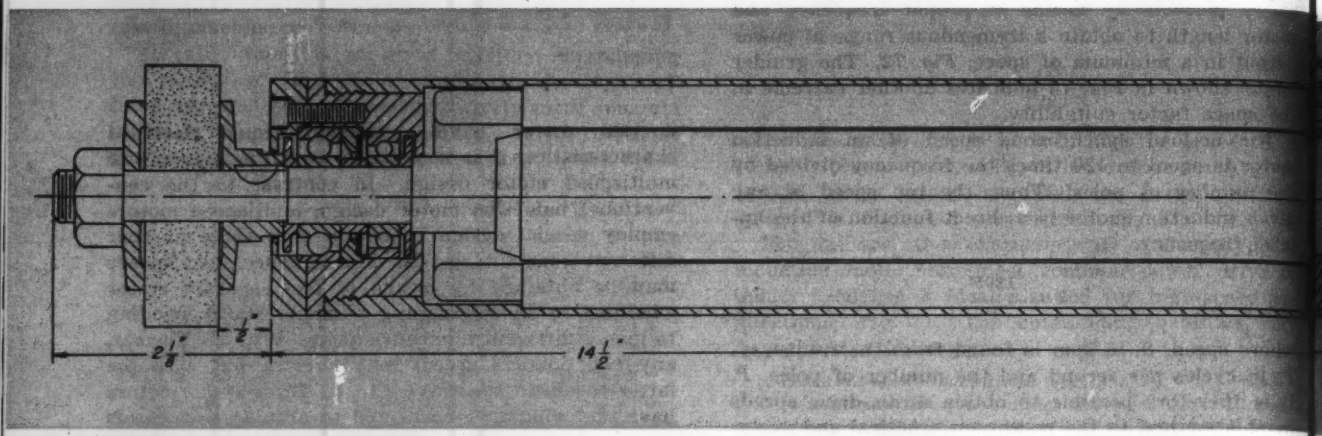


Fig. 73—Above—Another extreme in high-cycle design, this motor for fine internal grinding develops 1.5 hp at 15,000 rpm on 250-cycle current

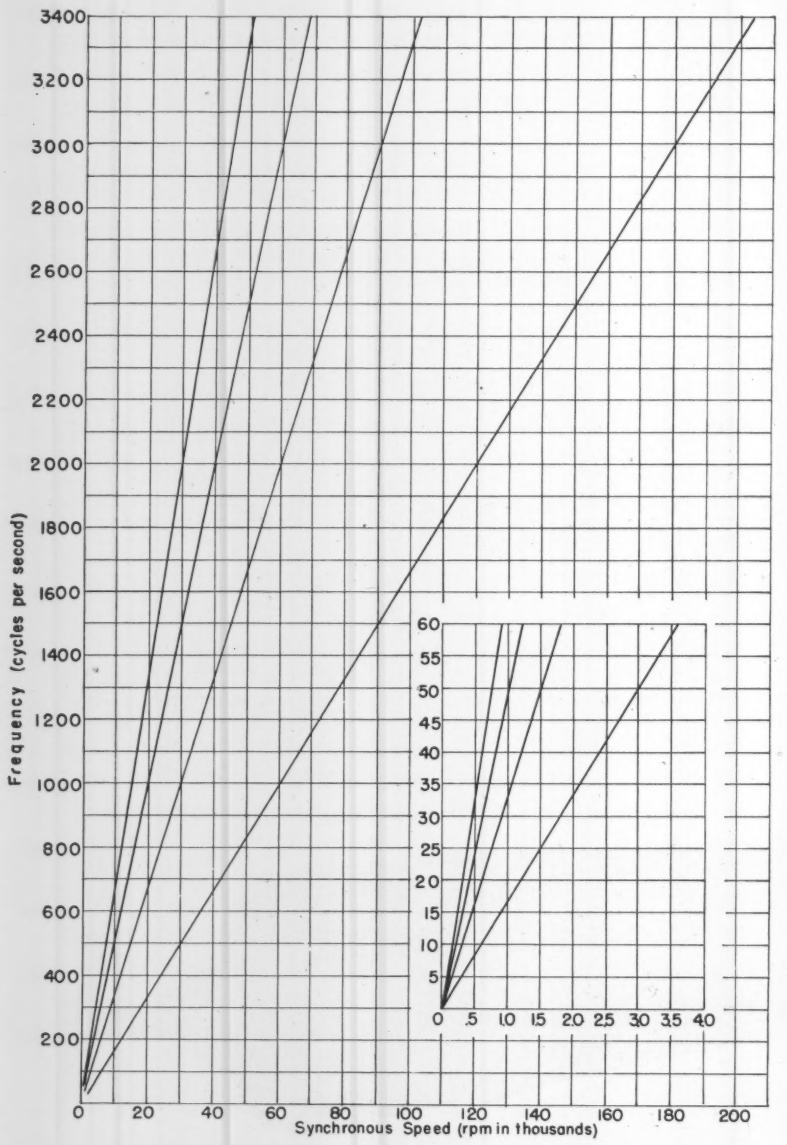


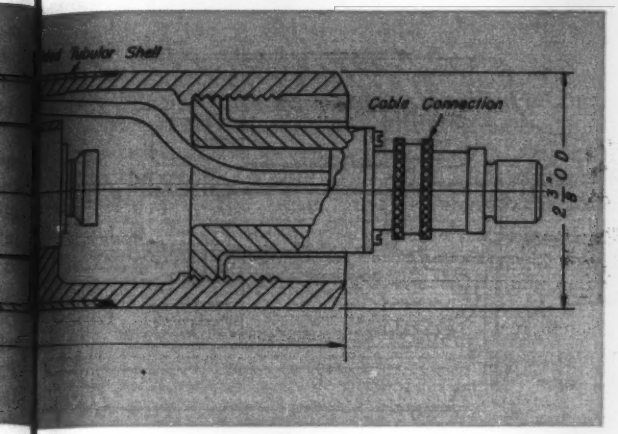
Fig. 74 — Left — Chart showing the speed-frequency characteristics of high-cycle motors. Actual load speed is slightly lower than the free synchronous speed

stant-torque design, as with the regular induction motor, being most generally applicable. Multi-speed motors can effectively reduce the cost of the power source owing to the possibility of using a set frequency machine.

On some applications of greater than 3600 rpm speed it is possible to employ stock squirrel-cage shell motors. Generally, however, such motors can be utilized satisfactorily only on frequencies up to 180 cycles. Over 180 cycles, the horsepower drops off rapidly owing to iron losses and heating problems. The horsepower increases directly with the frequency—adequate ventilation being assumed, of course—up to 180 cycles but beyond, the rating becomes:

$$hp = \left(\frac{F - 180}{120} + 3\right) hp_{60}$$

where frequency, F, is in cycles per second and developed horsepower is a function of the horsepower developed at 60 cycles. From this relationship it can be sen that stock shell motors are not advisable for use on the higher frequencies.



Generally speaking, the efficiency which may be achieved with any given weight of copper and iron increases with the applied frequency. However, the increase in efficiency is not rapid since it is possible to achieve motor efficiencies in the neighborhood of 80 per cent on 180 cycles. With the proper winding in the same given weight of materials, it would be possible to raise the efficiency to perhaps 85 or 90 per cent by increasing the frequency.

Heat Dissipation a Problem

Another important problem which confronts the designer is that of dissipating the heat which is generated in the relatively small mass. Since horsepower per pound increases with operational speed, it therefore creates the added problem of dissipating losses from an ever increasing area.

DESIGN: Successful design with high-cycle motors involves careful consideration of the overall machine requirements or "to-be-met" specifications. Of these, maximum horsepower-to-weight ratio, restricted diameter, abnormal speed, and speed variations are but a few. A special machine application might stress high speed and small housing diameter with overall length and weight of no importance. Again, space factor, diameter and length may be combined to provide maximum intermittent horsepower output with the weight factor overlooked.

Increasing the speed of an induction motor into the range of high frequencies is seldom an electrical problem. It is largely one of mechanical design and some of the major factors which must be recognized and coped with are as follows:

1. Proper speed-horsepower combination. Any specific horsepower at a specific speed can be selected. However, inasmuch as power output of a specific motor varies in direct proportion to the speed, a problem arises where variable speeds or a series of speed steps are desired. Usually it is necessary to limit the range of speed of any one motor and employ additional motors to obtain the desired power output at

Table 11—Frequency-Voltage Combinations for General-Purpose Applications

Cycles Per Sec	NEMA Voltage Range 1	2	3	4	5	6
60	110					
120	220	110				
180	330	165	110			
240	440	220	146			
360	330	220	110			
480	440	293	146			
600	367	183				
720	440	220	110			
900	275	138				
1080	330	165				
1440	440	220	110			
1680	256	128				
1860	284	142				
2040	312	156				
2880	440	220				

Fig. 75—High-cycle motor for internal grinding which develops 3 hp at 120,000 rpm on 2000-cycle current. Rear motor powers lubricating pump

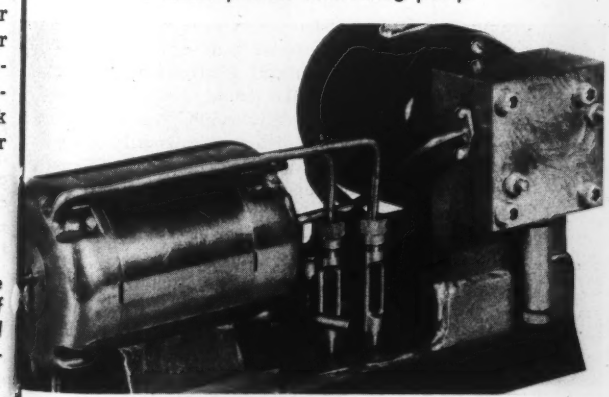
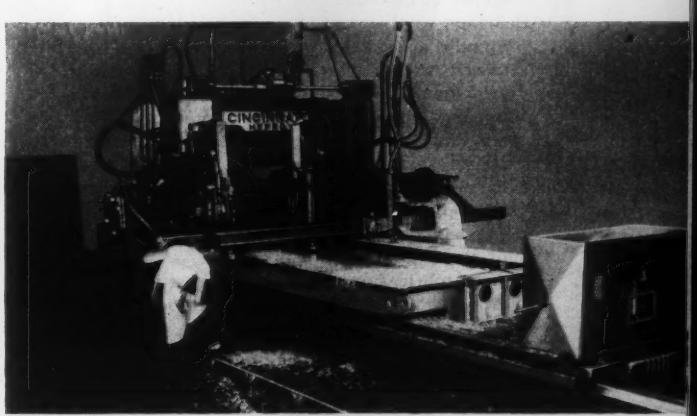


Fig. 76—Direct drive milling motors on this taper-skin mill are 50 hp, 3600 rpm, 440-volt, 60 cycle. Single-point cutters, 12 inches in diameter are fed at rates up to 84 inches per minute



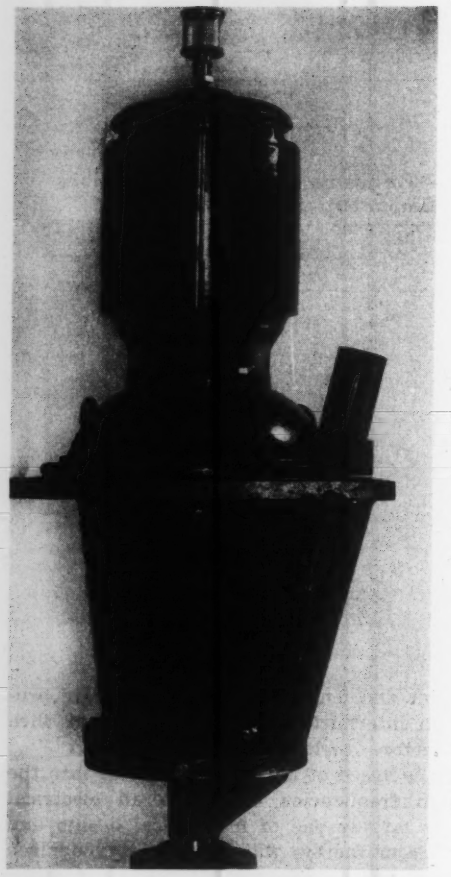


Fig. 77—Typical centrifugal dryer powered by 10,000 rpm high-cycle motors ranging from 5 to 15 hp

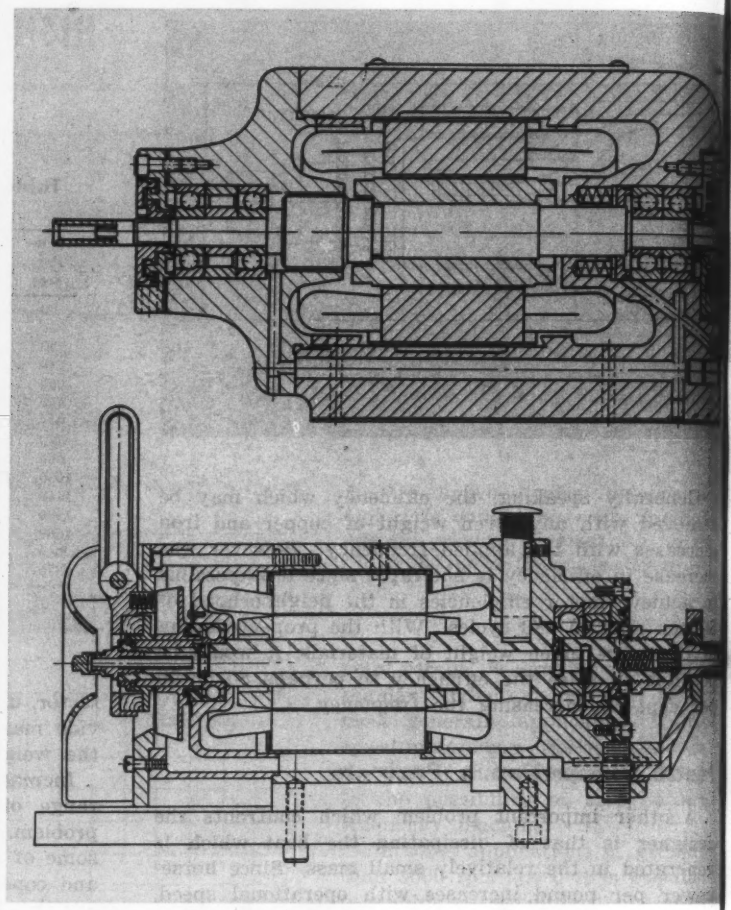


Fig. 78—Above—Combination radial router and drill motor of 5 hp, 250 cycle, 15,000 rpm, 220/440 volt, 3-phase type. Note special lubricators for bearings. Fan cooling of stator is employed

Fig. 79—Top—High-frequency Heald grinder wheelhead for 30,000 to 43,200 rpm speed. Water cooling of stator is utilized

maximum and minimum speed setting

2. Rotor designed for length rather than for diameter to obtain the necessary torque. Horsepower output varies directly with the stack length on a specific motor diameter, i.e., increasing the stack length one-third produces a horsepower increase of one-third
3. Bearings to withstand the desired load and speed. Spindle speeds up to 200,000 rpm have been attained with superprecision ball bearings properly mounted. Speeds up to 120,000 rpm are not uncommon, *Fig.* 75. Bearing manufacturers are now in a position to furnish ball bearings to operate at 120,000 rpm at a temperature of 250 F with an average life of 500 hours. However, large bearings with high speed and heavy loading will usually require specially designed types of sleeve bearings
4. Proper lubrication system for high-speed bearings. Air-oil mist lubrication for antifriction bearings is imperative at high speeds. Oil bath or splash lubrication cannot be tolerated in designs where bearing speed factors* over 200,000 are employed
5. Accurate balancing of moving parts. Because developed forces increase as the square of the speed, dynamic balance at high speeds is extremely important. Balancing equipment sensitivity** must be many times greater than that ordinarily required. Spacing of bearings must be such as to prevent shaft whip at critical speeds
6. Adequate means for maintaining proper operating temperature. Needs will depend upon whether the loading is cyclic or continuous. Fan cooling, *Fig.* 78, or some other means is normally a requirement to maintain optimum motor temperature. Where high horsepower and/or high speeds are involved, water cooling proves most effective, *Fig.* 79. Where extreme cases have been encountered and cost was no factor, the expedient of a refrigerant gas expanded against the rotor and stator through a hollow motor shaft has been effectively utilized, *Fig.* 80.
7. Means for arresting spindle coasting. A suitable brake for stopping spindle rotation may be im-

*"Superprecision Ball Bearings," by Thomas E. Rounds, MACHINE DESIGN, November, 1948 through May, 1949.

**For details on balancing sensitivity see "Determining Balancing Accuracy," by Werner I. Senger, MACHINE DESIGN, December, 1949, Page 113.

perative. Also, where cutters are mounted on the spindle a suitable locking device may be desirable.

It is interesting to note that in the design of high-speed spindle arrangements, inbreathing of contaminated atmosphere must be avoided. Partial vacuums frequently develop within high-speed shaft housings and a low-pressure air-oil mist is effective in preventing such inbreathing during or following shutdown, thus preserving bearing accuracy and extending spindle life. The action of the air-oil mist also helps cool the bearings.

As motor size and horsepower increases the mechanical problems are accentuated. Special bearing design as well as special consideration of the air resistance to rotor speed may become necessary, *Fig.* 81. The problems are seldom those of producing the necessary frequency inasmuch as the number of poles utilized in the high-frequency source equipment may be increased to permit operation at a relatively slow speed.

HIGH FREQUENCY SOURCES: The final consideration in designing for built-in high-cycle motors is the methods for producing current over the standard 60-cycle supply frequency. These sources can be roughly divided into two categories: (1) Power sources for a set frequency, and (2) power sources for a variable frequency.

Set Frequencies: As power sources for set frequencies, the frequency changer or converter and the alternator (motor-generator set) are used. The frequency changer is the most common owing to simplicity and low cost and is preferred in the smaller units under 50 kva, *Fig.* 82. The changer is virtually

Fig. 80—Right—Special high-cycle motor in which both water cooling of the stator and refrigerant gas cooling of the rotor is used

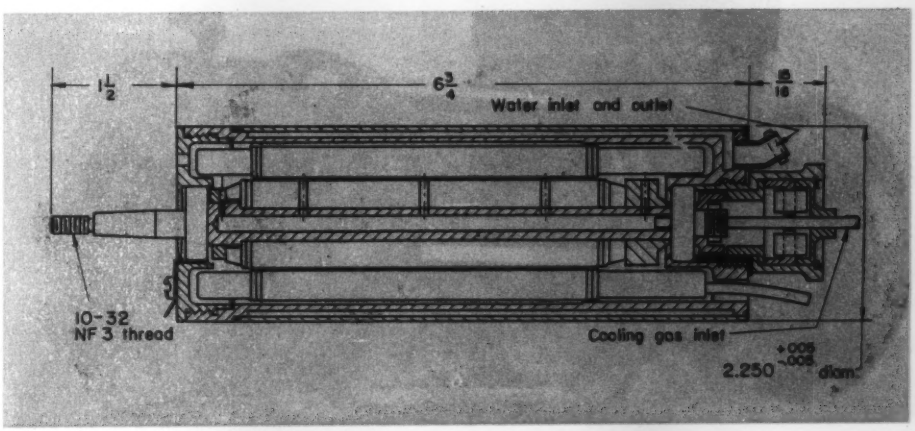


Fig. 81—Below—A special 3-phase induction high-frequency motor of 320 hp and 16,000 rpm. Motor is water jacketed and oil for high-pressure motor sleeve bearings is also passed over the stator coils. A vacuum pump lowers the interior pressure

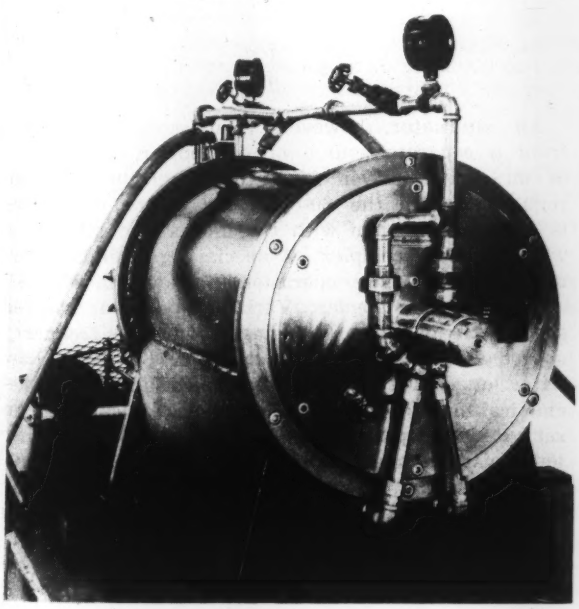
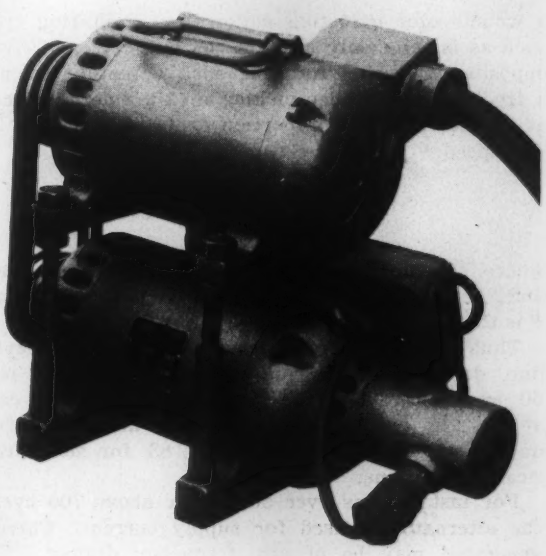
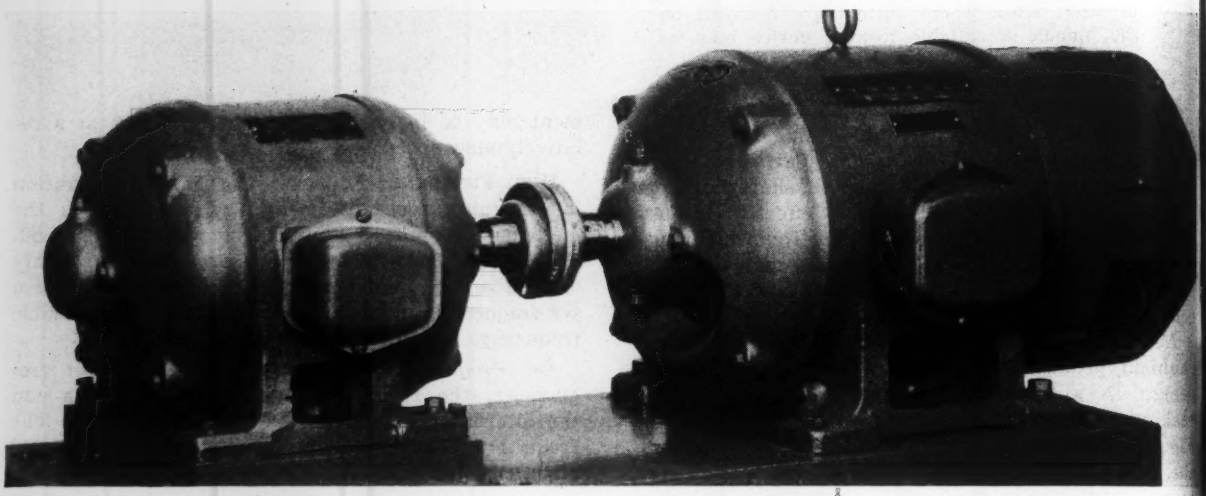


Fig. 82—Below—A 2 kva portable frequency changer





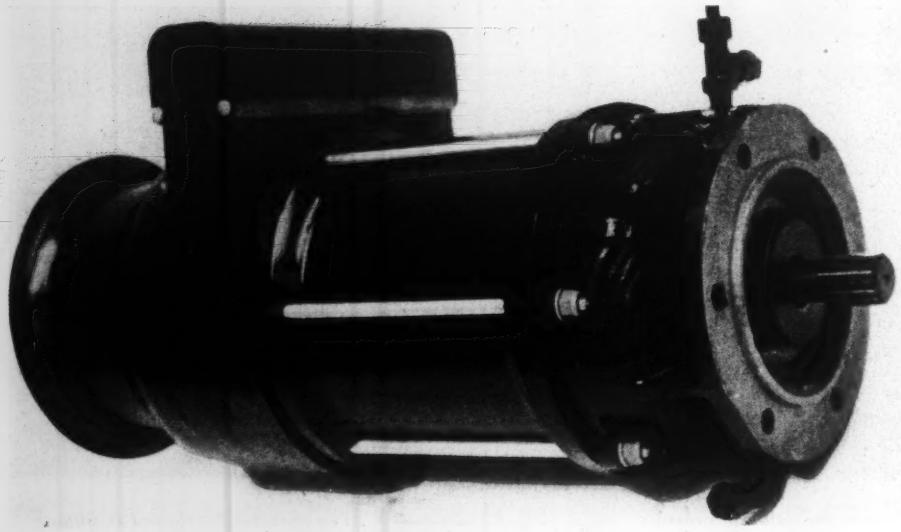


Fig. 83—Above—Direct coupled frequency converter. This type is built in ratings from 2 to 25 kw with frequency range of 120 to 700 cycles per second

Fig. 84—Left—A special 400-cycle lightweight alternator. Weighing but 52 pounds, this unit produces 62.5 kva

a wound-rotor induction motor of the slip-ring type such as is commonly used on elevator drives. Driven opposite to normal rotation, such a motor becomes a frequency multiplier picking up the line frequency and increasing it in proportion to its number of poles and speed

$$F = \frac{S\,P}{120} + 60$$

where output frequency, F, is in cycles per second and the speed, S, is in revolutions per minute and P is the total number of poles.

Thus an 8-pole frequency changer on a 60-cycle line, driven at 3600 rpm, would produce 240 plus 60 cycle, or 300-cycle current. Changers or converters, in general, are limited to an output frequency of 600 to 700 cycles, *Fig.* 83, for most practical applications.

For installations over 50 kva or above 700 cycles the alternator is used for supply current. Current generated may be of any frequency desired

$$F = \frac{P\,S}{120}$$

An alternator, however, requires d-c excitation from a separate source of d-c power, a rectifier, or an exciter driven from the generator. Voltage regulation with the 60-cycle a-c excited converter is much better than with the d-c excited alternator which is more complex. A special lightweight alternator for 400 cycle operation is shown in *Fig.* 84.

Variable Frequencies: Variable speeds or stepped speeds are obtained by using a variable frequency source. Varying the driven speed of a typical alternator or converter is accomplished either by mechanical means or electrically. A mechanically driven set of 2 kva capacity is shown in *Fig.* 85. With other units, various speeds are obtained by changing pulleys. Frequency changers or alternators driven by mechanical variable-speed belt arrangements are generally limited to a maximum output of 25 hp.

Units driven by electrical or other means for accurately controlling and varying the speed of the unit

Fig. 85—Above—A portable frequency changer with variable output. Mechanical variable-speed drive of this type is limited to about 25 hp safe maximum output

age and line loss may be large.

It is possible, however, to obtain differing speeds from one set frequency source by utilizing motors with differing numbers of poles. Likewise, with one frequency it would be possible to obtain several speeds with motors having multiple sets of poles. It must be remembered, however, that motor balance and running characteristics of bearings set for one speed seldom produce the same characteristics at another speed. For instance, a spindle balanced dynamically at 3600 rpm may not be in balance at 20,000 or 30,000 rpm. Heat dissipation, too, becomes a problem since at 3600 rpm and 1 hp, there are only 746 watts to dissipate, while at 14,400 rpm and approximately 4 hp there are some 2600 watts.

With these general problems confronting the selection of source and motor, it should be helpful to pre-

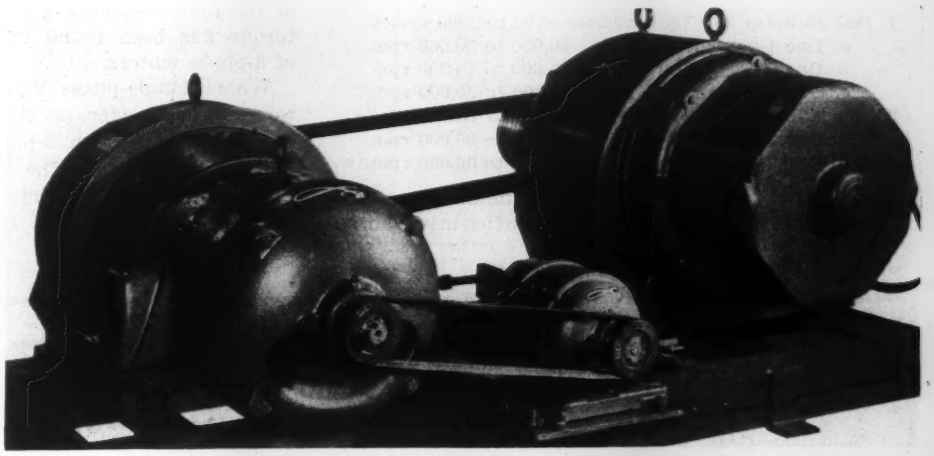
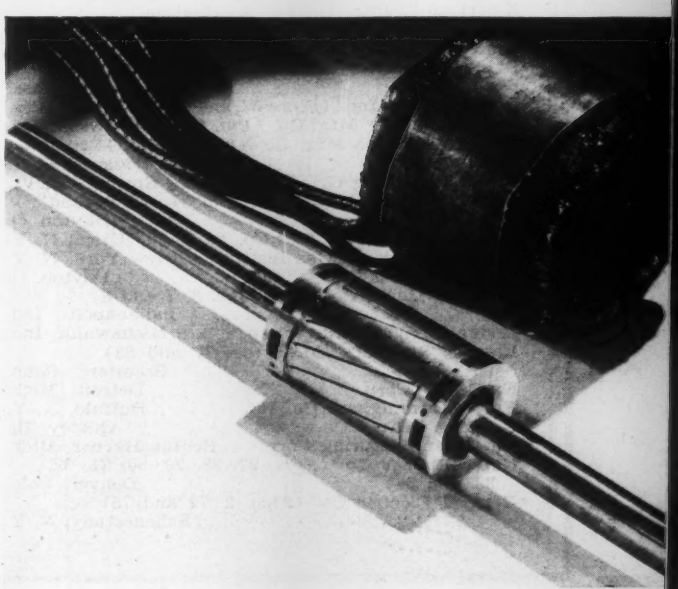


Fig. 86—Right—Special variable frequency alternator rated at 25 to 50 kw, 60 to 120 cycles. Direct-current variable speed motor drives both alternator and exciter

Fig. 87—Right, below—Special 2-phase high-cycle grinder motor shell and rotor. Rated at 2 hp at 36,000 rpm, the motor can be operated at speeds up to 100,000 rpm



virtually have no limitations within the practical range. *Fig.* 86 shows a 25 to 50 kw alternator for producing 60 to 120-cycle frequencies for a 25 to 50 hp, 3600 to 7200 rpm milling spindle motor.

Selection of Source, Motor and Frequency Combination: The constant-speed simple converter is the least expensive power source and offers satisfactory results. All other methods require more equipment, added control units and increase the mechanical problems.

Regardless of the method used it is generally most satisfactory to drive each high-cycle motor by means of a separate changer or alternator. Thus, each drive may be operated at the optimum desired speed. Also, where groups of spindles operate from one source, all are affected by source troubles and, too, unless all operate the major portion of the time, volt-

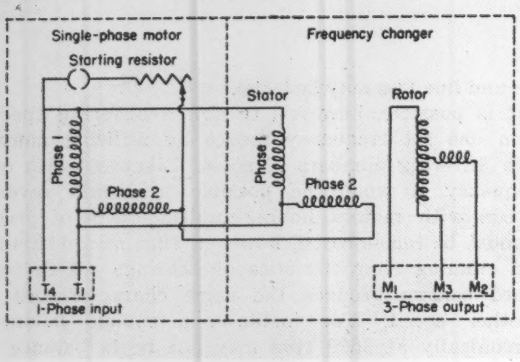


Fig. 88—Circuit for a special slip-ring converter which permits operation of 3-phase, 400-cycle motors from single-phase 60-cycle current

sent the solution to the difficult problem of establishing the most desirable setup for grinding spindles to cover the range from 20,000 to 90,000 rpm:

1. One 10 kw or 13.3 hp converter—700 to 1000 cycles
   a. One 4-pole motor ..... 20,000 to 30,000 rpm
   b. One 2-pole motor ..... 35,000 to 45,000 rpm
   c. One 2-pole motor ..... 45,000 to 60,000 rpm
2. One 1.5 kw or 2 hp converter—1000 to 1500 cycles
   a. One 2-pole motor ..... 60,000 to 80,000 rpm
   b. One 2-pole motor ..... 80,000 to 90,000 rpm

These motors, of course, develop maximum power at maximum speed but the range is sufficiently short to maintain the drop-off within useable limits.

Few standards have been set in the high-cycle drive field to date although moves are beginning in that direction. NEMA has established voltage and frequency standards for equipment over 60 cycles. In TABLE 10 these basic standards are listed. Power hand tools on the market today have already reached the 360 cycle speed range in the effort to reduce the weight per horsepower delivered. Because the voltage varies directly with the frequency, *Fig.* 70, it is apparent that voltage limitations are imperative to avoid hazardous installations. Consequently, the old standard of 1.22 volts per cycle has been superseded by a variable volts-per-cycle factor with 110 volts as the low and 440 volts as the high. A chart developed around the NEMA general-purpose standard is shown in TABLE 11. Here it can be seen that it is possible to generate the same frequency at two and in some cases three different voltages.

The majority of applications made to date are of the most common 3-phase motor design. However, some 2-phase and single-phase work has been done, *Fig.* 87. Use of 2-phase hookup simplifies the design of the high-frequency generator for supply. Starting torque has been found to be comparable with that of 3-phase motors.

Where single-phase supply is imperative, an interesting converter of the slip-ring type has been developed to furnish 3-phase, 400-cycle current for high-speed motors. The converter transforms the two-phase, 400-cycle current into 3-phase by means of a Scott connection as shown in *Fig.* 88.

MACHINE DESIGN is happy to acknowledge with much appreciation the invaluable co-operation of the following organizations in the preparation of this extensive article:

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American Pulley Co. ..... Philadelphia, Pa.
Baldwin-Duckworth Div., Chain Belt Co. ..... Springfield, Mass.
Berry Motors Inc. (*Figs.* 67 and 68) ..... Corinth, Miss.
Black & Decker Mfg. Co. (*Fig.* 71) ..... Towson, Md.
Brown & Sharpe Mfg. Co. (*Figs.* 9, 10, 11 and 13) ..... Providence, R. I.
Bryant Chucking Grinder Co. ..... Springfield, Vt.
Buckeye Tools Corp. (*Figs.* 49, 54, 63, 64 and 65) ..... Dayton, O.
Chicago Pneumatic Tool Co. (*Fig.* 59) ..... Utica, N. Y.
Compressed Air & Gas Institute ..... New York, N. Y.
Dayton Rubber Co. ..... Dayton, O.
Diamond Chain Co. Inc. (*Figs.* 6, 7 and 8) ..... Indianapolis, Ind.
Dodge Manufacturing Corp. ..... Mishawaka, Ind.
Electric Specialty Co. (*Figs.* 77 and 88) ..... Stamford, Conn.
Ex-Cell-O Corp. ..... Detroit, Mich.
Farrel-Birmingham Co. Inc. ..... Buffalo, N. Y.
Gardner Denver Co. ..... Quincy, Ill.
Gast Manufacturing Corp. ..... Benton Harbor, Mich.
Gates Rubber Co. (*Figs.* 27, 28, 29, 30, 31, 32, 33 and 34) ..... Denver, Colo.
General Electric Co. (*Figs.* 2, 72 and 75) ..... Schenectady, N. Y.
L. H. Gilmer Co. ..... Philadelphia, Pa.
B. F. Goodrich Co. ..... Akron, O.
Goodyear Tire & Rubber Co. ..... Akron, O.
Ingersoll-Rand Co. ..... Phillipsburg, N. J.
Keller Tool Co. (*Figs.* 62 and 66) ..... Grand Haven, Mich.
Leece-Neville Co. (*Figs.* 60 and 61) ..... Cleveland, O.
Leland Electric Co. (*Fig.* 84) ..... Dayton, O.
Link-Belt Co. (*Figs.* 5 and 12) ..... Indianapolis, Ind.
Merco Centrifuge Co. (*Fig.* 18) San Francisco, Calif.
Morse Chain Co. (*Figs.* 3 and 4) ..... Ithaca, N. Y.
National Broach & Machine Co. (*Fig.* 43) ..... Detroit, Mich.
National Electrical Manufacturers Association ..... New York, N. Y.
North American Aviation, Inc. (*Fig.* 76) ..... Los Angeles, Calif.
Onsrud Machine Works (*Figs.* 23, 24, 50, 51, 52, 53 and 78) ..... Chicago, Ill.
Pioneer Rubber Mills ..... San Francisco, Calif.
Quaker Rubber Corp. ..... Philadelphia, Pa.
Reliance Electric & Engineering Co. ..... Cleveland, O.
Republic Rubber Div., Lee Tire & Rubber Co. ..... Youngstown, O.
Russell Manufacturing Co. (*Figs.* 19, 20, 21 and 22) ..... Middletown, Conn.
Saco-Lowell Shops (*Fig.* 35) ..... Biddeford, Maine
Scandinavia Belting Co. ..... Newark, N. J.
Sharples Corp. (*Figs.* 25, 26 and 57) ..... Philadelphia, Pa.
A. O. Smith Corp., Motor Division (*Figs.* 73, 80, 81, 82 and 85) ..... Los Angeles, Calif.
Westinghouse Electric Corp. (*Figs.* 14, 36, 39, 40, 44, 45, 46, 47, 48, 69, 79 and 87) ..... Pittsburgh, Pa.
Wright Aeronautical Corp. ..... Wood-Ridge, N. J.

Chart Aids Design of Axial-Flow Turbines

By F. L. Schwartz
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University of Michigan
Ann Arbor, Mich.

DESIGN of axial-flow gas turbines is not essentially different from the design of steam turbines. Pressure ratios and volume changes of the expanding fluid are much less in a gas turbine but the fundamentals of design are the same even to the shape of the blade.

In any turbine utilizing a compressible fluid three design equations must be satisfied, involving three variables which give the designer some choice in selecting a suitable design. These equations and a chart for solving them are presented in this data sheet.

First design equation gives the pitch-line velocity, u, of the blades:

$$u = \frac{2\pi(1000N)r_m}{12(60)} = 8.74\ Nr_m \quad \ldots\ldots (1)$$

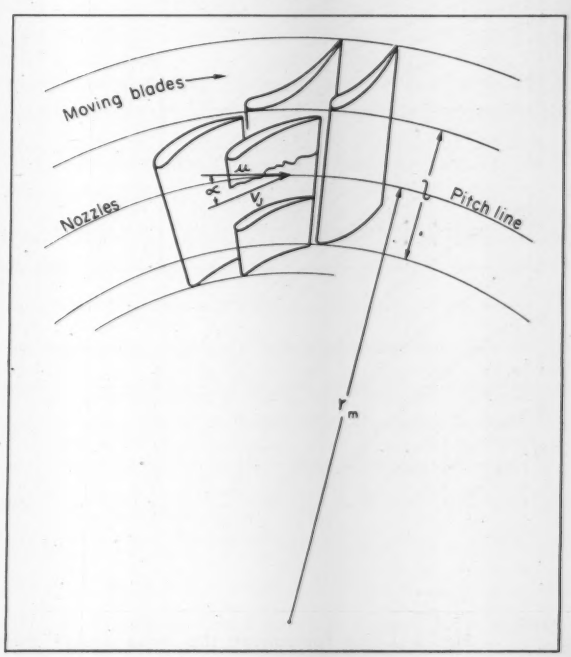
where N = speed of rotation, 1000 rpm; and r_m = pitch-line radius of blades, inches (see *Fig.* 1). The lower right-hand quadrant of *Fig.* 2 presents a graphical solution of Equation 1. The pitch-line velocity may also be expressed in terms of the fluid jet velocity, V:

$$u = \sigma V_j \quad \ldots\ldots (2)$$

where σ is a ratio which depends on the type of blading chosen. For maximum efficiency, the ratio $\sigma = u/V_j$ should be approximately 0.5 for an impulse wheel and 1.0 for a reaction wheel. Actually σ should equal $\frac{1}{2}\cos\alpha$ for the impulse wheel and $\cos\alpha$ for the reaction wheel, where α is the nozzle angle (angle between jet and direction of rotation). Since α is usually between 12 and 24 degrees the values of σ are close to 0.5 and 1.0, respectively.

For a given enthalpy drop per stage the ratio of the wheel speed for a 50 per cent reaction wheel to an impulse wheel will be $\sqrt{2}$. However, reaction stages usually have smaller enthalpy drops per stage than impulse stages, so the pitch-line velocity of the reaction wheel is equal to or less than the pitch-line velocity of the impulse wheel. Having determined the

Fig. 1.—Portion of turbine wheel showing basic dimensions and steam or gas velocities



Turbines

Data Sheet (Turbines)

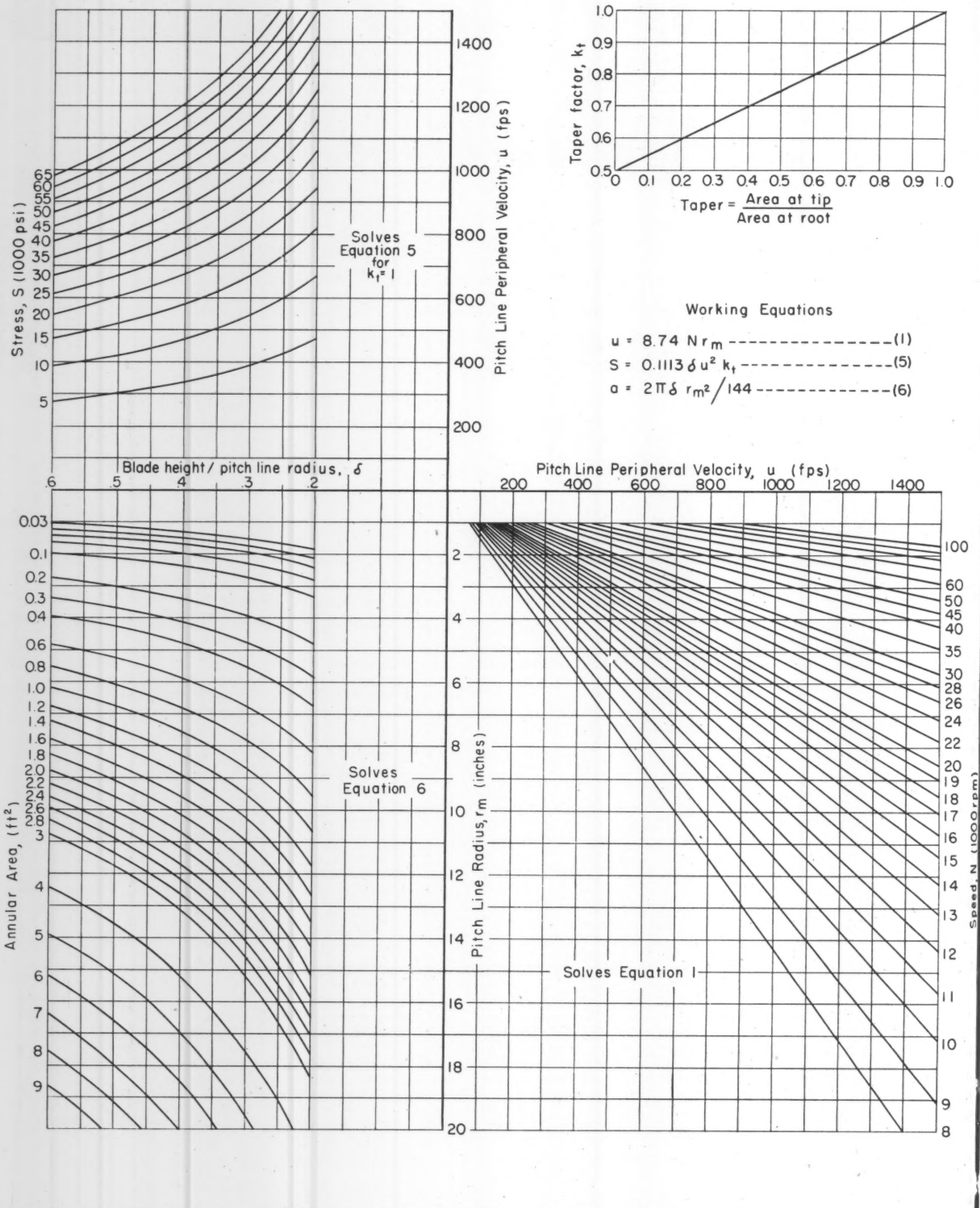

Stress, S (1000 psi)
Pitch Line Peripheral Velocity, u (fps)
Solves Equation 5 for $k_t = 1$
Taper factor, k_t
Taper = Area at tip / Area at root
Working Equations
$u = 8.74\ N r_m$ ----------(1)
$S = 0.1113\ \delta u^2 k_t$ ----------(5)
$a = 2\pi\delta\ r_m^2 / 144$ ----------(6)
Blade height / pitch line radius, δ
Pitch Line Peripheral Velocity, u (fps)
Annular Area, (ft^2)
Solves Equation 6
Pitch Line Radius, r_m (inches)
Solves Equation 1
Speed, N (1000 rpm)

Fig. 2—Chart for solving the three design equations required in calculating basic turbine dimensions

number of stages, type of blading, turbine pressure ratio and inlet temperature, the pitch-line velocity u becomes a fixed design factor. Most gas turbines use a type of blading which is designed for impulse at the root and varying degrees of reaction along the length of the blade with 20 to 40 per cent reaction at the tip.

Second equation gives the stress, S, at the blade root:

$$S = 28.4wN^2lr_mk_t \quad \ldots\ldots (3)$$

where w = density of blade material, lb per cu in.; l = blade length, inches (see *Fig.* 1); and k_t = taper factor of the blade.

The stress in a blade is maximum at the root and results almost entirely from the centrifugal force in the rotating blade. Except in slow-speed turbines the bending stress in the blade is negligible compared to the centrifugal, and is ignored in the present analysis. If desired this refinement can be made after the initial design is selected and the design modified to suit the refinement. However, it is questionable whether the properties of the materials are well enough known and the fabrication accurate enough to warrant a high degree of refinement in stress calculations.

If the value of w for Vitallium ($w = 0.3$) is substituted, and the blade length l is expressed as a fraction of pitch-line radius, $l = \delta r_m$, Equation 3 becomes

$$S = 8.5\delta N^2r_m^2k_t \quad \ldots\ldots (4)$$

Inasmuch as the pitch-line velocity of the blades is given in terms of r_m and N in Equation 1, Equation 4 may be written

$$S = 0.1113\delta u^2k_t \quad \ldots\ldots (5)$$

The upper left-hand quadrant of *Fig.* 2 represents a convenient graphical solution of Equation 5 for taper factor $k_t = 1$. The taper factor is a function of the degree of taper, which may be defined as the ratio area at tip to area at root, assuming uniform taper.

As Equation 3 indicates, the stress varies directly with the density, illustrating the desirability of using low-density blade materials. For example, comparing a tungsten blade with a blade made of titanium carbide, the ratio of densities is about 3 to 1 so the tungsten must have better than three times the strength of titanium carbide to compete on equal terms.

Third design equation gives the area at the exit of the blade, *Fig.* 1:

$$a = \frac{2\pi r_m l}{144} = \frac{2\pi\delta r_m^2}{144} \quad \ldots\ldots (6)$$

where a is the area, in square feet, which must accommodate the required fluid flow. Equation 6 may be readily solved with the aid of the chart in the lower left quadrant of *Fig.* 2.

The mass rate of flow and specific volume leaving the stage can be calculated after the horsepower, pressure ratio, temperature and assumed efficiencies are set. The annulus area is equal to the volume rate of flow divided by the axial velocity. The axial velocity may be determined by the vector diagram for the blading or may be assumed. For gas turbines the axial velocity varies from 300 to 800 fps, the higher values being used in jet propulsion units and the lower values in power units.

The three design equations involve the seven quantities u, δ, r_m, S, a, N and k_t. The quantities u and a are predetermined by the size, blade type, cycle, number of stages, etc. The taper factor k_t is a designer's choice and depends on the type of blading and flow area required. Of the four remaining quantities three are determined by the equations and one must be chosen. Usually the speed, N, is determined by the desired speed of the compressor if it is a gas turbine unit or by the output element driven by the turbine. If the speed is not predetermined the stress is at least limited by the choice of blade material. Hence any set of values for the three remaining quantities which satisfy the three equations are satisfactory within certain limits.

Design Values Quickly Determined

With the six quantities, all except k_t, of the three equations plotted on a chart as shown in *Fig.* 2, a choice of design values may be determined quickly and with precision. The stress lines in the upper left quadrant are for an untapered blade. For a tapered blade the root stress will be the stress for the untapered blade times the taper factor found from the auxiliary diagram in the upper right quadrant. For satisfactory design δ should lie between the limits of 0.2 and 0.6.

In addition to aiding the selection of design dimensions, the chart shows some interesting results of turbine design. For example, in a small gas turbine the gas flow is low and the annular area is low. The chart shows that under these conditions the radius must be small (unless partial admission is used) and the speed must be very high. The ratio δ has little effect on the radius at small radii but a large effect on the radius at large radii. Slow-speed turbines of reasonable diameter can only be had at low values of u and this results in many stages of low enthalpy drop per stage.

The steps to be followed in a design would be, given horsepower, pressure ratio, inlet temperature, and assumed efficiency:

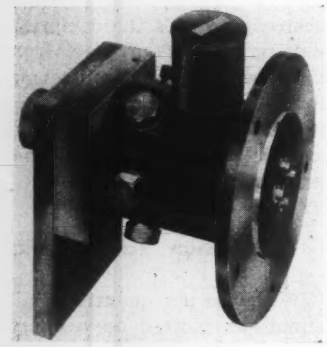
1. Select number of stages
2. Calculate V_J
3. Choose type of blade and calculate u
4. Calculate volume flow rate leaving last stage
5. Assume an axial velocity with the aid of blading vector diagram and calculate annulus area a
6. Choose degree of taper and read taper factor from chart in upper right quadrant of *Fig.* 2
7. Select limiting stress and calculate equivalent untapered blade stress
8. Read ratio δ from chart
9. Read radius r_m from chart
10. Read speed N from chart.

If the resulting values are not satisfactory, modify δ and S to give a more desirable design.

NEW PARTS AND MATERIALS

For additional information on these new developments see Page 211

Current Collector Head



This collector head transmits high currents from stationary to rotating member. Standard unit is rated 10,000 amp at continuous 12-volt direct current at 400 rpm; heads with different ratings can be supplied. Resistance is less than 10 microhms. Overall dimensions are 10 x 11½ x 13½ in., with face plate diameter of 10 in. Head incorporates internal built-in lubrication pump and face plate for mounting bus bars for current connection. Collector head utilizes Timken tapered bearings and is self-supporting from flange faceplate mounted on rotating member. Shunts, which connect stationary member, prevent head from rotating. Contact surfaces consist of silver contacts permanently fused to copper segments and held by constant spring pressure against rotating shaft. Manufacturer: Sciaky Bros. Inc., 4915 W. 67th St., Chicago 38, Ill.

For additional information circle MD 1 on Page 211

Miniature Radial Ball Bearings

Designated as the RO series, these annular, miniature ball bearings are made in the same twelve sizes as the R series precision bearings. Sizes start at 0.0295-in. bore and 0.1181-in. OD. Furnished with ball retainers, the bearings have no filling slots and hence are suitable for both radial and thrust loads. They are usable for applications where high precision is not a prime factor and where space saving and economy are desired. Manufacturer: Landis & Gyr Inc., 104 Fifth Ave., New York, N. Y.

For additional information circle MD 2 on Page 211

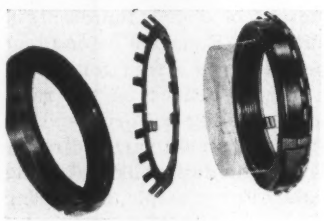
Cold-Working Stainless Steel

Work-hardening much slower than conventional 18-8 types, No. 10 chrome-nickel austenitic stainless steel is especially suitable for fabrication of parts such as fasteners that require drastic cold heading or upsetting. Since the steel remains nonmagnetic after severe cold coining and extrusion, it also has application in making of parts for precision instruments. Type analysis is 16 per cent chromium, 18 per cent nickel and maximum of 0.08 per cent carbon. The steel resists a wide variety of organic and inorganic chemicals and foodstuffs but is not resistant to hydrochloric and other halogen acids. Material can be cold or hot worked, welded and machined satisfactorily. After hot-working operations, corrective anneal by water quenching is necessary to prevent intergranular corrosion.

Typical mechanical properties after water quenching from 2000 F: Yield strength, 35000 psi; tensile strength, 75000 psi; elongation in 2 inches, 55.0 per cent; reduction of area, 72.0 per cent; brinell hardness, 145; Rockwell hardness, B 80. Forms available include forging billets, hot rolled bars and forgings, cold drawn and ground bars, and wire and wire rods. Manufacturer: Carpenter Steel Co., Reading, Pa.

For additional information circle MD 3 on Page 211

Bearing Locknuts and Lockwashers



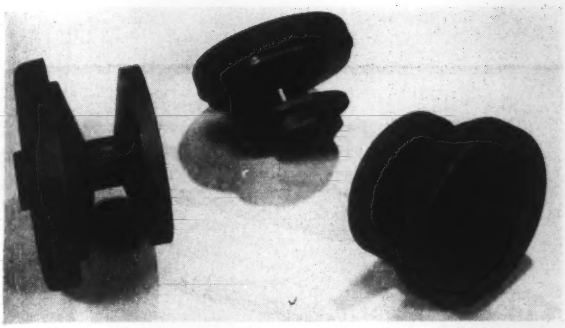
Effective locking of ball and roller bearings in correct position on shafts of 5/16 to 7 11/16-in. diameter is afforded by these locknuts and lockwashers. Conforming to ABEC and SAE standards, the units are installed and removed easily, rust-resistant, vibrationproof and burr-free. Manufacturer: Link-Belt Co., 307 N. Michigan Ave., Chicago 1, Ill.

For additional information circle MD 4 on Page 211

Automatic Clutch Pulley

Particularly adaptable for applications requiring high starting torque, this variable-pitch pulley clutch automatically tightens the belt, operating equally well in either direction. Clutch pulley engages at smaller diameter than that at which it drives so that the unit operates as combination clutch and automatic transmission.

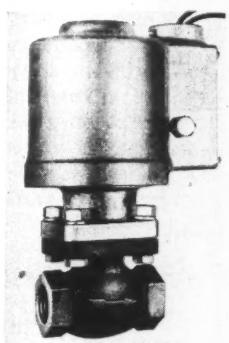
Pressure caused by belt tension forces the movable half away from the fixed half of the pulley. Wedge weights actuated by centrifugal force move radially outward along the inner conical surface to force it axially toward the fixed half, thereby causing pres-



sure on V-belt. These weights wedge between mechanism cover and floating cone. A garter spring reacts against the centrifugal force of wedge weights to withdraw their force as speed is reduced. Cover, support tube, wedge weights, floating cone, and end flange rotate as a body. Unit is available in sizes for ½, ⅝ and ¾-in. shafts; and special sizes or types can be obtained. Manufacturer: Light Inspection Car Works, P. O. Box 186B, Hagerstown, Ind.

For additional information circle MD 5 on Page 211

AC-DC Solenoid Valve



Precision seating of this solenoid valve for hard-to-handle liquids and gases provides tight shutoff and trouble-free operation. Solenoids are hand wound, moistureproof and oversized to insure high ratio of power to friction. Laminated core keeps operating current to minimum. Heavy valve body is noncorrosive metal and one-piece valve stem and disk is stainless steel. Plunger tube is hermetically sealed. Valve body is joined to bonnet with flange and gasket to provide tight seal, yet can be taken apart easily without distorting valve body and seat. Power supply can be either ac or dc. Valve remains in open position until current is cut off. Manufacturer: Charles Engelhard Inc., 850 Passaic Ave., E. Newark, N. J.

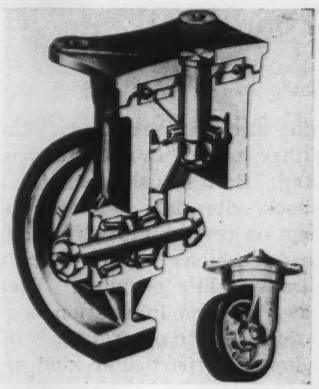
For additional information circle MD 6 on Page 211

Electronic Relay System

Extreme sensitivity in industrial control applications is afforded by this electronic relay system which utilizes cold cathode type gas tubes. Mounting on standard 4-in. electrical connection box, unit operates from 115-v 50-60-cycle line. Operation is initiated by contact of drop wire, contact-making instrument or any conductive medium between two contacts having resistance as high as 1 million ohms. Extreme sensitivity allows operation of load relay with silver contacts and high contact pressure which allow for control with load currents as high as 4 amp, dependent upon voltage and load inductance. Load relay contacts are arranged to permit choice of either opening or closing a circuit, or simultaneously opening one circuit and closing another. 'Hold' circuit is incorporated for optional use so that load relay can remain energized from a momentary initiating medium or contact. When 'hold' circuit is in use, reset switch on cover will manually restore circuit to another operation cycle. Manufacturer: Servo-Tek Products Co., 4 Godwin Ave., Paterson 1, N. J.

For additional information circle MD 7 on Page 211

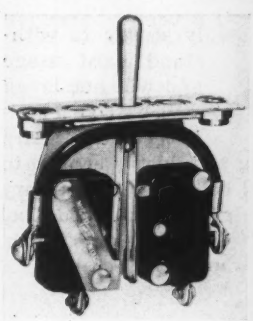
Heavy-Duty Wheels and Casters



Available with both rigid and swivel casters, these heavy-duty industrial trailer wheels are cast aluminum alloy and are designed to minimize clutch wear, battery strain and mechanical fatigue on power towing equipment. Wheels and casters can be obtained in four different mountings: caster mount, inside mount, outside mount and stub axle outside mount. Features include sensitive swiveling action, lifetime sealed-in lubrication and permanently adjusted 1¼-in. chrome molybdenum steel kingpin. Timken or Bower tapered roller bearings of largest possible size in relation to specified wheel diameter are used to provide easy starting and smooth rolling. Manufacturer: Aerol Co., 2820 Ontario St., Burbank, Calif.

For additional information circle MD 8 on Page 211

Momentary Contact Switch

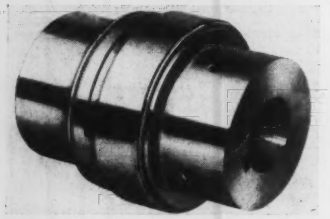


Three-position 1AT1 momentary contact switch assembly employs two double-throw switches to increase number of possible circuits that can be switched. Unit consists of metal bracket in which self-centering toggle lever and two type V3 switches are mounted. One switch is normally closed and the other normally open, with jumper wire connecting common terminal of two switches. Operating toggle lever to right closes normally-open circuit of one switch, and moving toggle lever to left opens normally-closed circuit of the other switch. In either direction from center position, there is a 25-degree total travel of operating lever. Electrical rating of

switches in assembly is 1-amp 125 to 250-volts ac or 1-amp 28-volts dc noninductive load, and 6-amp 28-volts dc inductive load. Weight of unit is approximately 2 oz. Manufacturer: Micro Switch, Freeport, Ill.

For additional information circle MD 9 on Page 211

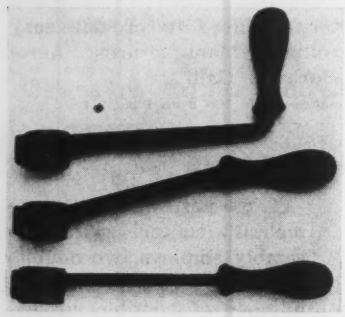
Flexible Couplings



Ease of installation and larger bore capacities characterize the improved Types C and H couplings designed for use on 50 to 1000-hp machinery. Inside sleeve is made in two parts to provide sleeve in each body. Collar can be installed on either half of coupling and is held in place by snap-ring. Both models instantly adjust for misalignment, shock, vibration, oscillation, surge or backlash. Except on reversing load, half of the cushions are idlers; load cushions can be quickly interchanged without disassembling coupling. There is no wear on metal jaws, and no lubrication is required. Couplings are available with several types of cushion materials, each adapted to particular kind of service, indoors or out. Manufacturer: Lovejoy Flexible Coupling Co., 5001 W. Lake St., Chicago 44, Ill.

For additional information circle MD 10 on Page 211

Malleable Iron Machine Handles

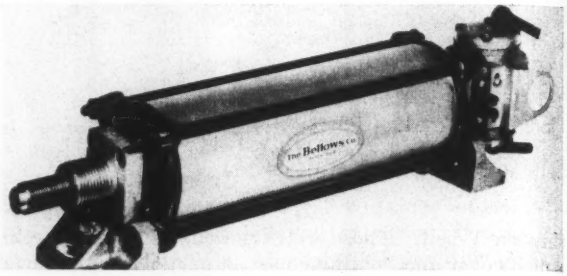


Crank, straight and offset styles of malleable iron machine handles are available in various sizes ranging from 4 to 18 in. They are sufficiently sturdy to withstand hard usage and will not break if accidentally struck sharp blow. They are free machining and ductile enough to permit user to make additional bends when necessary to suit his requirements. Manufacturer: Reid Tool Supply Co., 709 Baker St., Muskegon Heights, Mich.

For additional information circle MD 11 on Page 211

Thrust-Type Air Motor

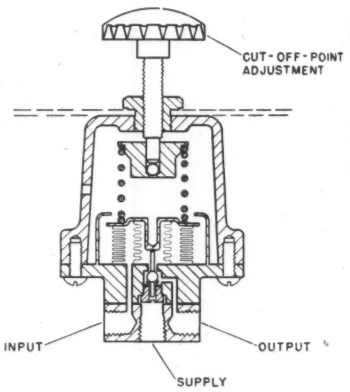
Model BM2 air motor with a bore of 1¾ in., develops thrust in pounds approximately twice that of the applied air line pressure in psi. It is obtainable with built-in air valve either manually controlled or air powered. Speed controls for regulation of piston



rod speed in either or both directions are an integral part of both valves. Unit is available in almost any required stroke length. Manufacturer: Bellows Co., Akron, O.

For additional information circle MD 12 on Page 211

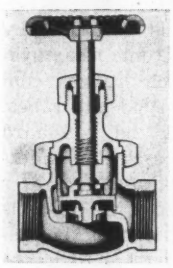
Pneumatic Relay



Providing sharp cut-off action, this pneumatic relay is suitable for use with equipment, such as air-electric switches, requiring accurate high-speed operation. Supply pressures as high as 50 psi may be used. Cut-off point can be set for any pressure between 0 and 35 psi; overloading up to 70 psi will not damage instrument. Design provides high amplifying ratio so that small change in input pressure results in full change in output. When used with high magnification type Moore pneumatic comparator gage and commercial pressure switch, this relay will operate signal light on dimension change as small as 0.000002-in. Manufacturer: Moore Products Co., H & Lycoming Sts., Philadelphia 24, Pa.

For additional information circle MD 13 on Page 211

Liquefied Petroleum-Gas Valves

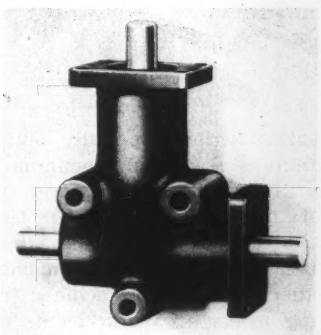


This line of globe, angle and check valves has been approved by Underwriters' Laboratories for use on liquefied petroleum-gas service at pressures to 250 psi. All three types have screwed ends and are available in sizes from ¼ to 2 in. To insure tight seating, valves have composition disk which is cemented into disk holder. Tightness at stuffing box is afforded by one-piece molded asbestos packing. In addition to above-

mentioned service, valves can be used for other applications such as hot or cold water, compressed air, gasoline and light oils. Manufacturer: Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.

For additional information circle MD 14 on Page 211

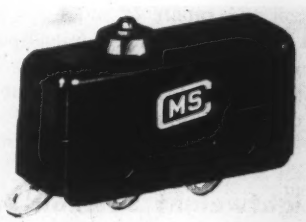
Geared Angle Drive



This improved model ANGLgear standardized right - angle bevel gear drive, suitable for use in either manual or power operated systems includes rectangular four - hole mounting flange with internal pilot on both ends of housing. This mounting arrangement is in addition to three-hole side mounting originally employed. Unit is modified further to make available 2:1 reduction as optional feature in addition to standard 1:1 ratio. Standard unit is rated at 250 lb-in. static torque capacity or to transmit 1/3-hp at 1800 rpm continuous duty. If required, units can be furnished with 500 lb-in. static torque rating. Manufacturer: Airborne Accessories Corp., 25 Montgomery St., Hillside 5, N. J.

For additional information circle MD 15 on Page 211

Miniature Snap-Action Switch



Measuring 49/64 x ⅜ x ¼-in., this precision switch has 'pin' plunger actuator, solder lug type terminals and single-pole double-throw circuit arrangement. Lightweight unit is enclosed in plastic case and is rated 5 amp at 115 to 230 volts ac and 2 amp at 28 volts dc. Maximum operating force required is 7 oz, and minimum release force is 1 oz. Maximum movement differential is 0.03-in., maximum pretravel is 0.008-in. and minimum overtravel is 0.005-in. Manufacturer: Micro Switch, Freeport, Ill.

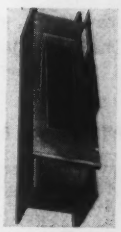
For additional information circle MD 16 on Page 211

Lightweight Expanded Metal

Made of 22-gage metal, Minimesh expanded metal has 3/16 x ½-in. diamond pattern mesh. It can be produced in carbon steel, copper, aluminum or other metals. Material can be used for small screens and grilles where light weight and ornamental effect are desired. Manufacturer: Penn Metal Co., 250 E. 42nd St., New York, N. Y.

For additional information circle MD 17 on Page 211

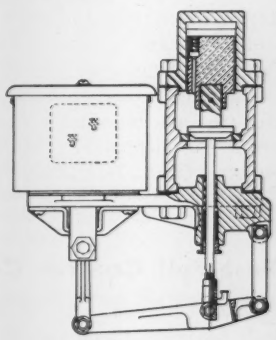
Lighting Panelboards



Model NMO-XX circuit breaker lighting panelboards are designed to fit into web of 8-in. wide-flange structural beams. Box is constructed with outward flanges to allow full unobstructed space on each side of the interior for wiring purposes. Top and bottom wiring gutters measure 5 in., and side gutters, 1⅛ in. Cable trough and pull boxes can be supplied for concealing the main and branch circuit wires running from panelboard to ceiling or roof trusses. Units are available for both single-phase three-wire and three-phase four-wire services with 15, 20, ,30, 40 and 50-amp branch circuits. Thermal-magnetic circuit breaker units that plug-in on cylindrical bus bars are employed. Manufacturer: Square D Co., 6060 Rivard St., Detroit 11, Mich.

For additional information circle MD 18 on Page 211

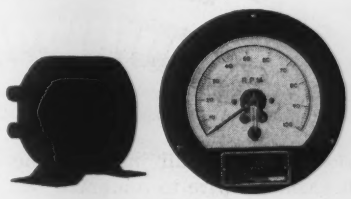
High-Velocity Solenoid Valve



This series 4000 direct-operated solenoid valve employs integral dashpot to end water hammer that can result from sudden valve closing in high velocity pipe line. Designed for normally closed operation, valve slows closing action by 1½ to 2 seconds. Furnished in ¾, 1, 1¼, 1½ and 2-in. sizes, it handles temperatures up to 400 F, and smaller sizes will operate under differential pressures up to 150 psi. Standard unit has cast-iron body and bronze piston, or all-bronze construction can be furnished. Valve and seat are of stainless steel. Manufacturer: Johnson Corp., Three Rivers, Mich.

For additional information circle MD 19 on Page 211

High-Speed Electric Tachometer

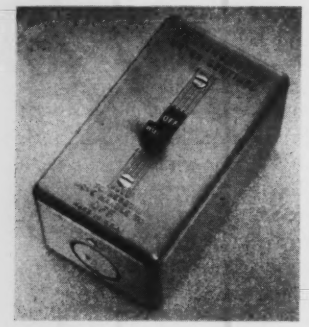


Featuring flush mounting indicator with 250 - degree scale, model 12-M electric tachometer is accurate within 2 per cent of full scale deflection. Scale length is 8.7 in., diameter of case is 5½ in. Standard scales are calibrated in revolutions per minute, but special types are available to read feet per minute, gallons per minute, impressions per hour, or any other unit of speed. Design of 12-pole permanent magnet generator insures steady pointer action even

at low speeds. Unit can be obtained with base, flange or SAE mounting and is suitable for continuous operation at speeds up to 10,000 rpm. Manufacturer: Chicago Electric Tachometer Co., Madison, O.

For additional information circle MD 20 on Page 211

Manual Motor Starter

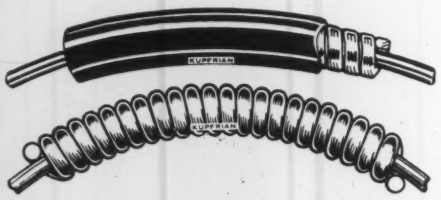


Especially suited to controlling and protecting motors to 1-hp rating, Motor Sentinel manual starter with positive bimetallic overload protection fits any standard 2½-in. wall box. With front-removable heater of proper rating, unit permits time delay for starting and allows motors to carry overloads up to their safe capacity without disconnecting them from line. Positive operation is assured by over-center toggle mechanism. Contacts are double-break silver, and parts are copper alloy or plated steel. For surface mounting, unit is available in alloy enclosure with Bonderized baked enamel finish. Special cast-iron enclosures are obtainable for watertight and explosion-proof applications. Manufacturer: Westinghouse Electric Corp., Box 2099, Pittsburgh 30, Pa.

For additional information circle MD 21 on Page 211

Push-Pull Control Casings

Added to the line of Monocoil casings for push-pull controls, these two types are specially adaptable to marine and corrosive applications. First type consists of high-strength bronze flat wire completely enclosed by water-tight vinyl cover. Flat-wire construction provides smooth inner surface for control wire, and solid wall construction affords high bend

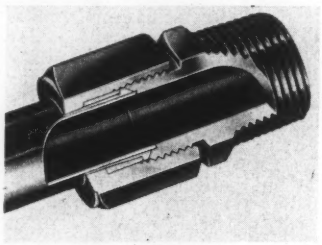


resistance. Small inside diameter of casing closely fits control wire to minimize 'lost motion' between the two parts.

Second type uses stainless steel wire, and has approximately double the strength of brass or bronze. Made primarily in ¼-in. OD, the casings safely withstand most chemicals. Manufacturer: Kupfrian Mfg. Co., 217 Prospect St., Binghamton, N. Y.

For additional information circle MD 22 on Page 211

Flareless Tube Fitting



This flareless tube fitting, adaptable to close-quarter connections, limits tube gouging and distortion and permits quick installation without special tools. Gripping action of fitting sleeve forms leakproof seal and makes fitting especially suitable for copper, steel, stainless steel and aluminum tubing. It is available in all standard shapes and sizes for tubing to 1-in. OD. Many special combination shapes and sizes to 2 in. can be furnished. All shapes such as elbows, tees, crosses, etc., are machined from forgings, all other parts being machined from bar stock. The fitting meets all requirements of JIC hydraulic standards for industrial equipment and specification AN-F-47. Manufacturer: Aircraft Fitting Co., 1400 E. 30th St., Cleveland 14, O.

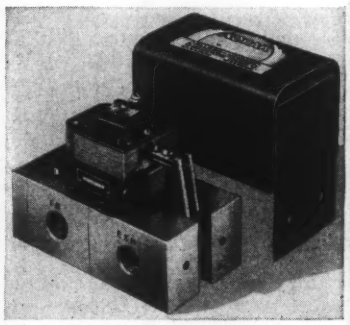
For additional information circle MD 23 on Page 211

Aluminum Welding Electrode

Developed for welding aluminum and aluminum alloys, Eutectrode 2101 welding electrode has a tensile strength of 35,000 psi. It deposits metal rapidly and smoothly with good color match and corrosion resistance. Obtainable in ⅛ and 3/16-in. sizes, the electrode is recommended for reclamation of new castings and repair of old ones. Weld puddle is visible at all times because fumes and smoke have been eliminated by improving electrode coating. Manufacturer: Eutectic Welding Alloys Corp., 40 Worth St., New York 13, N. Y.

For additional information circle MD 24 on Page 211

Lightweight Solenoid Valve



Compact, lightweight, ring - type valve embodies solenoid - actuated pilot rather than conventional direct pull of solenoid on the sealing mechanism. This Nopak Model SL valve handles air pressures from 40 to 250 psi; at 220 v, it consumes only 0.96-amp in developing pull of approximately 3 lb. Before solenoid is energized, line pressure is always on same cylinder port, with other cylinder port open to exhaust. When solenoid is energized, cycle is re-

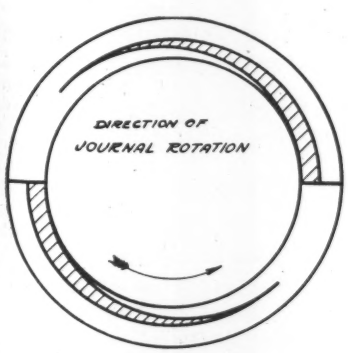
Flanged Sleeve Bearings

THE addition of a flange on a sleeve bearing often serves as a very useful device to the designer. For instance it is an ideal way of meeting the problem of end thrust. Or it can serve to locate or anchor the bearing; to act as a divider; or to form a recess to hold the lubricant. In some cases, particularly in electric motor bearings, the flange has a beveled edge to insure alignment.

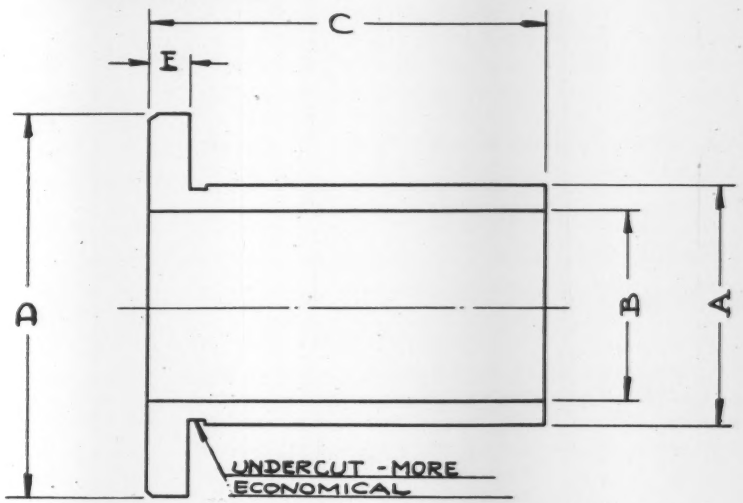
In cast bronze bearings the flange can be located anywhere on the outside diameter and can be made any thickness as required. In sheet metal bearings the location of the flange is limited to the ends and cannot be any thicker than the metal used in the body of the bearing.

In applications where the principal use of a flange is to meet a thrust condition it is well to include oil grooves on the face of the flange. In cast bronze the most popular type is the wedge shape as illustrated herewith. This permits the introduction of oil in order to produce the hydraulic pressure required to separate the flat surfaces. Sheet metal bearings usually employ a design known as "tear drop."

In applications where the operating speed is comparatively slow, it is often wise to use graphite impregnated bearings. The graphite can be included on both the inside diameter and the flange of the bearing. In some cases it is often difficult or impossible to locate the oil intake in the proper place to insure good lubrication while at the same time make it convenient and accessible. The use of flange in applications of this nature provides a reservoir for the lubricant. The oil intake can be placed in the most convenient place on the housing while the corresponding hole in the bearing can be placed where operating conditions demand.



Recommended type of groove on face of flange.



Measurements to specify when ordering Flanged Bearings.

Powder Metallurgy

When the application permits, the most economical method of producing flanged bearings is by Johnson LEDALOYL. With this method the complete bearing is formed, under pressure, to required size and tolerance. All machine work is eliminated and you gain the advantage of the self-lubricating action. Many sizes of LEDALOYL flanged bearings are standard stock items, ready for immediate delivery. Our new catalogue gives a complete listing of sizes available. Write for your copy.

How to order Flanged Bearings

When ordering or specifying flanged bearing it is important that *all* dimensions are given. This diagram can be used as a guide. When using flanged half bearings be sure to specify whether they are FULL halves or SAW CUT. Saw Cut bearings can be used where shims are included. Full cut bearings are more expensive but are necessary for precision fits. The use of an undercut under the flange lowers production costs but does not interfere with the efficient operation of the bearing.

A. Outside diameter of bearing.
B. Inside diameter of bearing.
C. Length—over-all—of bearing.
D. Outside diameter of flange.
E. Thickness of Flange.

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

●

This bearing sheet data is but one of a series. You can get the complete set by writing to—

Johnson Bronze
SLEEVE BEARING HEADQUARTERS
525 S. MILL ST. • NEW CASTLE, PENNA.

versed. Valve is energized by continuous duty, standard solenoid unit which is unaffected by oil.

Three-piece valve body is machined from drawn aluminum and is easily dismantled. Sealing mechanism consists of valve ring and Neoprene O-rings. Weighing 5¾ lb, unit measures 5⅞ x 3 x 5½ in. Valve is now built in ½-in. pipe size only, but it can be bushed down readily to ⅜ or ¼-in. pipe size. Subplate mounting is available as optional equipment. Manufacturer: Galland-Henning Mfg. Co., Milwaukee 15, Wis.

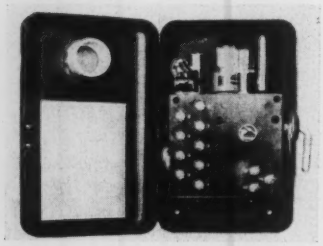
For additional information circle MD 25 on Page 211

Hydraulic Oil

Recommended for use under extremes of heat and cold, Gulflite Oil 5W for hydraulic motors, pumps and similar equipment has a viscosity of 90-100 SSU at 100 F. Oil body, heavy load-carrying capacity, and nonfoaming and free flowing characteristics are maintained throughout entire range of temperatures at which such equipment is normally operated. Made from high-grade solvent and paraffinic-type crude oils, the product is refined by the Alchlor process. It is compounded with additives which retard oxidation and increase its capacity to lubricate parts under heavy loads. Manufacturer: Gulf Oil Corp., Pittsburgh, Pa.

For additional information circle MD 26 on Page 211

Photoelectric Control

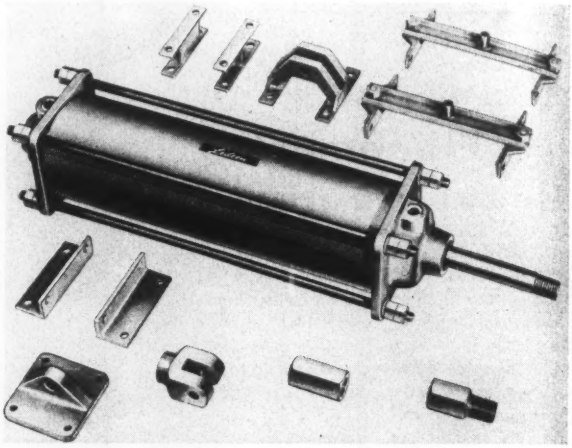


Having general machinery application, this series 20 photoelectric control is also recommended for counting or conveyor control, short-range signal systems, motor or valve control, stop-motion control, or other similar industrial process controls. It can be obtained with phototube integral with control or located in separate small housing for applications where space is limited. To make it useful in all types of safety applications, it is supplied with forward and reverse circuits so that tube replacement is indicated by either energized or de-energized relay. Terminal panel connections provide for operation from 115 and 230-v supply, and tamperproof sensitivity adjustment permits positive operation over varying distances between phototube and light source. Manufacturer: Photoswitch Inc., 77 Broadway, Cambridge 42, Mass.

For additional information circle MD 27 on Page 211

Air and Hydraulic Cylinders

Medium-duty series of air and hydraulic cylinders is applicable to remote control operation, and wherever straight line motion is required. Tie-rod construction



gives positive protection against leakage at joint of tube and head, and chevron packing seals piston rod. Synthetic cups and automatic cushion are standard. Various head and rod attachments for these standardized cylinders provide almost universal mounting requirements. Manufacturer: Ledeen Mfg. Co., 1600 S. San Pedro St., Los Angeles 15, Calif.

For additional information circle MD 28 on Page 211

Flexible Stainless Steel Hose

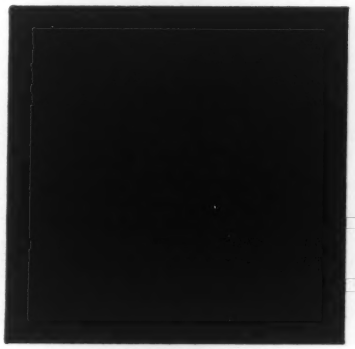
Fabricated in sizes from 5/16 to 6-in. ID in single or multiple ply construction, corrosion-resistant, flexible stainless steel hose is available for working pressures to 2000 psi. Wall thicknesses range from 0.003-in. in single ply to 0.03-in. in multiple ply construction. Hose is supplied with helical or annular cor-

rugations and with or without braid covering, depending upon pressure and other factors of service use. Open or closed corrugation pitch types are offered to meet varying requirements of flexing, cycles of vibration and misalignment. Manufacturer: Chicago Metal Hose Corp., 1304 S. Third Ave., Maywood, Ill.

For additional information circle MD 29 on Page 211

Air-Hydraulic Solenoid Valve

Type DS Quick-as-Wink control valve is usable at air pressures to 200 psi and oil or water pressures to 250 psi. Valve employs a double solenoid traveling ⅛-in. to move a pilot valve plunger, applying air to operating piston which actuates main valve plunger. When solenoid is de-energized, pilot cylinder is exhausted through pilot valve, permitting main plunger



from under many hats

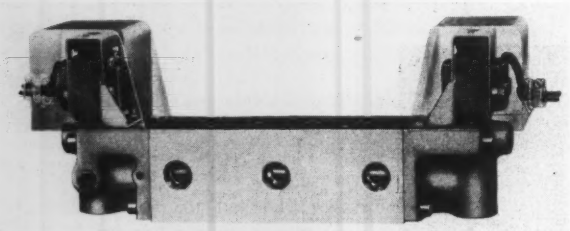
... the soft fedora of the salesman ... engineer, the furnace stained cap of the foundry craftsman ... the symbolic mortar-board of the scientist ... the eerie mask of the welder ... these symbolize the skills and traditions necessary to produce finer castings. Side by side, welded by years of experience, the men beneath these hats unite their individual skills to produce the quality and excellence that is known to all industry as a stamped Ⓢ on Sivyer high alloy, low alloy and carbon steel castings.

SIVYER

SPECIALISTS IN HIGH ALLOY AND SPECIFICATION STEEL CASTINGS

SIVYER STEEL CASTING COMPANY • MILWAUKEE Ⓢ CHICAGO Ⓢ

to return. Reduced current requirements obviate need for intermediate relays and simplify electric control circuits. Because solenoid plunger and pilot cylinder weigh only a few ounces and have short travel, valve can be operated at high speeds and normal air pressures. Housing of valve and all internal parts are

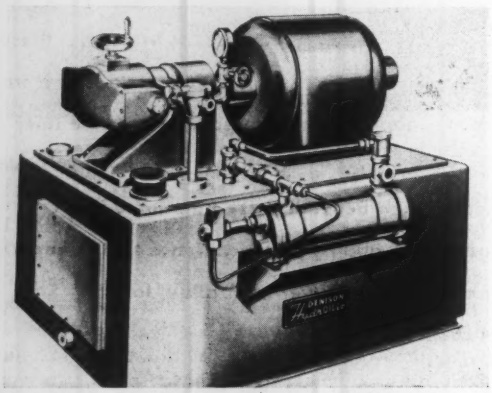


brass or bronze, while plungers and operating pistons are stainless steel. Sizes from 3/8 to 2 in. are furnished in two, three and four-way designs with solenoid control in both directions or automatic return in one direction. Manufacturer: C. B. Hunt & Son Inc., Salem, O.

For additional information circle MD 30 on Page 211

Oil-Hydraulic Pumps

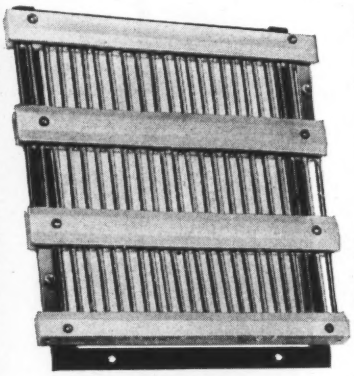
Available in 22 models with capacities from 9 to 35 gpm, these oil-hydraulic pumping units can be used for regulative pressures to 5000 psi. They are of the axial piston type and deliver either constant or variable volume. Each unit consists of reservoir base with all operating components including pump, relief valve, gages and electric motor drive mounted on re-



movable top cover. Welded steel reservoir is obtainable in capacities of 55, 110 and 165 gal. Variable volume pumps can be supplied with a choice of oil delivery controls—handwheel, pressure compensating, stem or cylinder. Motors range from 5 to 75-hp rating according to service requirements. Accurate control of pressure within range can be made with threaded body type relief valve. Heat-exchanger type oil cooler and vacuum gage to permit checking oil filter condition can be obtained as optional equipment. Manufacturer: Denison Engineering Co., Columbus 16, O.

For additional information circle MD 31 on Page 211

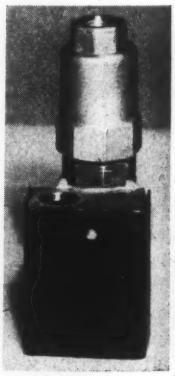
Heating Element Rack



Versatility, ease of installation and economical operation are provided by standard size heating element racks. Suitable for temperatures to 750 or 1350 F, they can be arranged horizontally or vertically and are available in ratings from 1.5 to 300 kw. Folded-and-formed continuous ribbon type elements are located in high temperature refractory porcelains, supported on 12-in. square steel or stainless steel frames. Drilled for mounting, racks can be used on kettles, autoclaves, plating tanks, core and armature bake ovens, air heaters, grilles, dryers, melting pots, salt baths, glass and wire annealing equipment and air conditioning units. Manufacturer: Trent Inc., 4573 Wilde St., Philadelphia 27, Pa.

For additional information circle MD 32 on Page 211

Flow Interlock Control Device



Responding to flow of cold water to open or close an electrical contact, flow interlock control device operates independently of water pressure. It closes contact when flow of water exceeds preset amount and opens when flow falls below preset amount. The device actually acts as a fuse in a circuit which depends upon water cooling for protection. Adjustment can be made to control circuit for any flow differential from 1 to 4 gal per min. Flow differential between cut-in and cut-out of electrical contact is 0.2-gal minimum and 0.4-gal maximum. Electrical rating of alternating-current device is 10 amp at 125 v, 5 amp at 250 v, and 3 amp at 460 v. It will withstand maximum line pressure of 125 psi. Mechanism is available complete with union fitting and strainer attachment. Interlock can control amount of water flow by installation of throttling or regulating valve ahead of device. Manufacturer: General Electric Co., Control Div., Schenectady 5, N. Y.

For additional information circle MD 33 on Page 211

This resilient tape makes it easy to

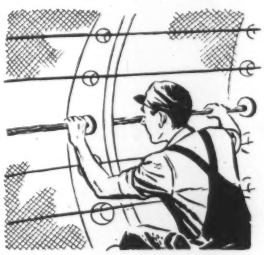
- **cushion adjacent parts**
- **eliminate squeaks**
- **simplify assembly**
- **make units non-skid**

One of the simplest ways to handle the jobs mentioned above is to apply Armstrong's DK-153 Tape. Quick and easy to use, it protects adjacent parts from rubbing, quiets squeaky joints, and makes slippery surfaces non-skid.

Armstrong's DK-153 Tape is a compressible, high-friction material, made of cork-and-sponged-rubber and backed with a pressure-sensitive adhesive. It is easy to apply. You merely strip the fabric backing from a piece of Armstrong's Tape and press into place on any clean, dry surface. For extra convenience in assembly, this tape can be supplied in the form of die-cut parts.

Whether you make diesel locomotives or pottery lamps, airplanes or dial telephones, there is probably a place in your product where Armstrong's DK-153 Tape can do a similar job for you. Or perhaps it can do an existing job, better and more easily.

In either case, we'd like a chance to demonstrate the capabilities of Armstrong's DK-153 Tape. For further information, samples, and prices write Armstrong Cork Company, Gaskets and Packings Department, 5104 Arch Street, Lancaster, Penna. Available for export.



CUSHION ADJACENT PARTS

Aircraft wiring, cushioned with Armstrong's DK-153 Tape at points where it passes through bulkheads and structural members, is protected from excessive chafing. Because this tape is compressible, it also prevents insulation crushing when positioning flanges are tightened.

ELIMINATE SQUEAKS

Strips of Armstrong's DK-153, applied to structural members in railway coaches and diesel locomotives, absorb relative motion and prevent sliding between panels and frame. Thus, it eliminates squeaks and prevents destructive abrasion between panels and framework.

SIMPLIFY ASSEMBLY

The pressure-sensitive adhesive backing on Armstrong's DK-153 Tape simplifies assembly for a pottery lamp manufacturer. Die-cut to the shape of the lamp bottom, pads of Armstrong's Tape are ready to use without the time-consuming hand gluing operation, previously required.

MAKE UNITS NON-SKID

To prevent dial telephones from slipping on polished surfaces, many local telephone companies now equip their phones with non-skid pads cut from Armstrong's DK-153 Tape. These pads may be applied at the factory or by the telephone serviceman at the point of use.

ARMSTRONG'S DK-153 TAPE

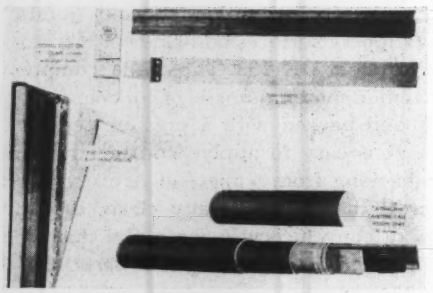
One of Armstrong's Cork-and-Rubber Products

ENGINEERING DEPARTMENT EQUIPMENT

For additional information on this new equipment see Page 211

Collapsible Drawing Board

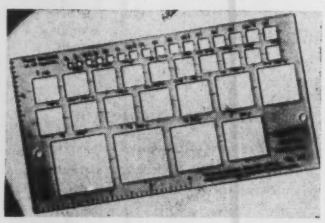
Features of this drawing board, straight edge and T-square unit are collapsibility, light weight and low cost. All parts of disassembled unit can be carried in 3 x 21-in. cylindrical carrying case. The support for the drawing is nonrigid plastic which is inserted



in notches provided on the straight edge. This support also holds drawing paper without use of thumbtacks. Precision-built T-square is in two parts so that rule proper can be separated from guide portion. Manufacturer: Frank E. de Beu & Co., 3502 Market St., Philadelphia 4, Pa.

For additional information circle MD 34 on Page 211

Drafting Template



Containing thirty squares measuring from 3/32 to 1⅜ in., No. 30 square template is designed for general drafting use. Edges of template are scaled to allow construction of squares to 4 in. Squares are in increments of 1/64-in. to 11/32-in., in 32nds to ¾-in., in 16ths to 1 in. and in 8ths to 1⅜ in. Each square size is clearly marked, and all have center guide line. Template is 0.030-in. plastic with matte finish. Manufacturer: Rapidesign Inc., Box 592, Glendale, Calif.

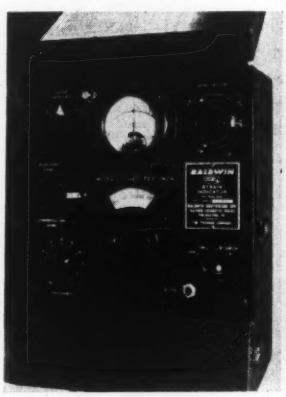
For additional information circle MD 35 on Page 211

Servo Development Apparatus

Development of servomechanisms, computers, etc., is facilitated by this flexible, mechanical development apparatus. Foundation of the unit is an 8 x 8 x ⅜-in. aluminum mounting plate containing 3/16-in. holes spaced ¼ in. apart. The plate can be mounted on legs to support it above a table, to separate it from similar plates, or to mount another plate at right angles. Precision-machined blocks for holding accessories, and bearing blocks to hold shafts, are available and can be bolted to the base plate in any desired location. A wide range of parts is offered including: stainless steel, centerless ground, 3/16-in. shafting in various lengths; box collets to locate shafts; and an assortment of stainless steel gears. Manufacturer: Servomechanisms Inc., Mineola, N. Y.

For additional information circle MD 36 on Page 211

Battery-Powered Strain Indicator



Designed for use with SR-4 bonded resistance wire strain gages, direct-reading Type L portable battery-powered strain indicator is a refinement of Type K. Four major improvements have been made: The ten 'thousands' steps are increased from 1000 to 2000 microinches per in. per step, giving a total of 20,000 microinches per in. This feature minimizes use of range extender switch without affecting accuracy; range extender has been increased to 20,000 microinches with an accuracy within plus or minus 50 microinches. This change extends balancing range of the instrument to plus or minus 30,000 microinches and permits measurements in plastic range of metals; the unit can be used with completely external wheatstone bridge circuit without modifying internal circuit; and an oscilloscope connector jack is provided for applications in rephasing problems encountered when using exteremely long leads between strain gages and instrument.

Instrument is designed and calibrated for 120-ohm gages with gage factors of 1.77 to 2.20. Gages of other resistances and gage factors can be used if necessary. Converter is available for operating indicator on 110-v 60-cycle single-phase power when required for continual operation. Manufacturer: Baldwin Locomotive Works, Philadelphia 42, Pa.

For additional information circle MD 37 on Page 211

MAKE YOUR MACHINES PRODUCE MORE, LAST LONGER AND COST LESS TO RUN

with Alemite Centralized Lubrication

Here are 4 tested ways to assure increased production, longer life and lower maintenance cost to *any* machine you design or build. Every Alemite Centralized Lubrication System eliminates "downtime" for lubrication, allowing 60 full minutes of peak production for every hour your machine is operating.

Dependable, automatic lubrication can be profitably applied to any machine—big or small, simple or complex. And it's the safest, surest way to eliminate human error, assuring your machines the exact lubrication *you know* they require. Alemite Centralized Lubrication cuts machine maintenance, "downtime" and repairs—*in advance!*

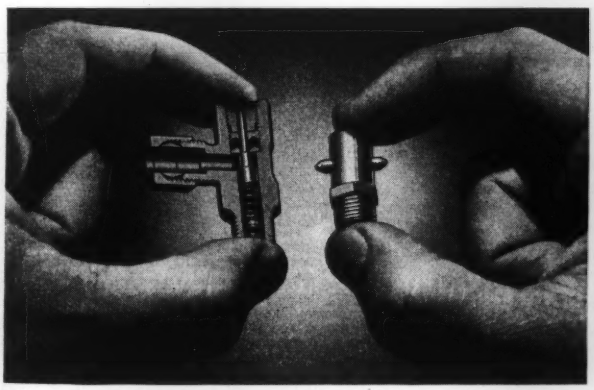
No *one* system, of course, meets the lubrication requirements of *every* machine. That's why Alemite perfected *four different systems* of centralized lubrication—each available with manual control or automatic control by the clock.

Your technical questions are invited by our staff of lubrication engineers. Any Alemite representative will gladly conduct an informative, desk-top demonstration of these systems at your convenience, without obligation. Catalogues on any Alemite system, above, are available upon request. Address Alemite, Dept. R-40, 1850 Diversey Pkwy., Chicago 14, Illinois.

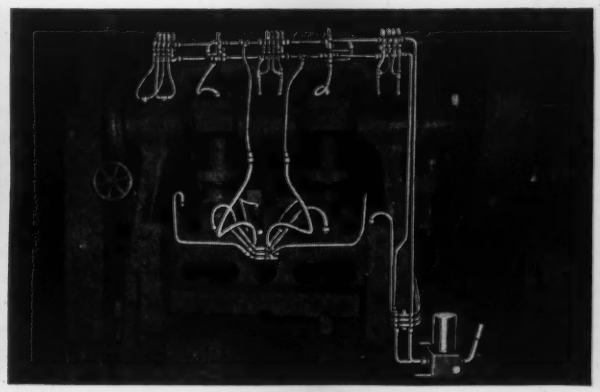
Another Product of STEWART WARNER

ALEMITE

Modern Lubrication Methods That Cut Production Costs



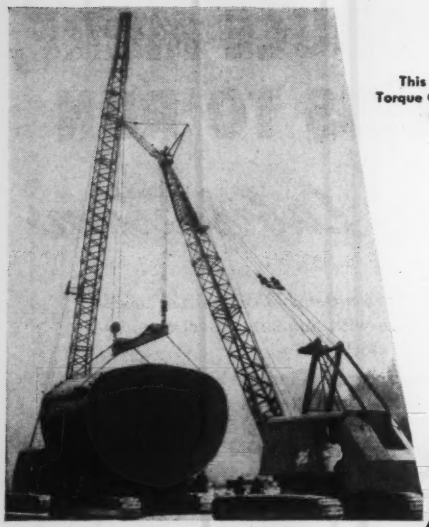
The famous Alemite "Midget" System—a hydraulic measuring valve for oil or grease, slightly larger than a pin-type fitting. Highly adaptable.



The Alemite Dual-Manifold System—heavy-duty hydraulic system for measured, volume delivery of oil or grease to *two* bearings per valve, during machine operation.

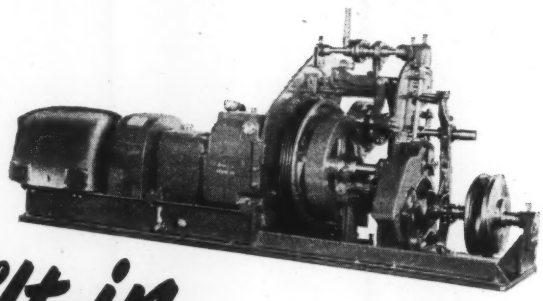
The Alemite Lubrometer System—a single-line, valve-metered system that delivers oil or grease to bearings while the machine keeps right on producing.

The Alemite Dual-Progressive System—a completely enclosed hydraulic system. Each valve measures oil or grease to *two* bearings while the machine operates.



This 50-ton Manitowoc Speedcrane uses a Twin Disc Torque Converter to deliver power for smooth gear-shift-free lifting operations.

This electric-motor driven Johnson and Bassett spinning mule incorporates a Twin Disc Hydraulic Coupling in addition to three Twin Disc friction clutches.

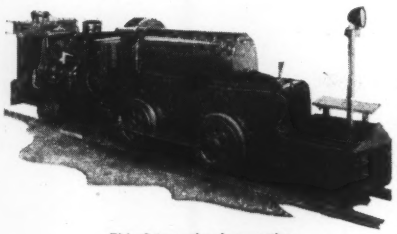


Built-in preventive maintenance in these units

More and more manufacturers of heavy-duty powered equipment are designing preventive maintenance right into their units with Twin Disc Hydraulic Drives—Couplings, Torque Converters and Power Take-offs.

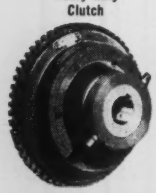
Hydraulic drives, by eliminating any direct connection between driving and driven parts, provide a cushion to absorb shock loads before they can become destructive. Vibration from engine or motor is dampened out. And these drives permit a smooth pick-up of the load from a dead stop—with no possibility of stalling out the engine or motor.

Preventive maintenance is an accomplished fact on Twin Disc Hydraulic Drive-equipped machinery. For complete information, write today for helpful literature. TWIN DISC CLUTCH COMPANY, Racine, Wisconsin (Hydraulic Division, Rockford, Illinois).

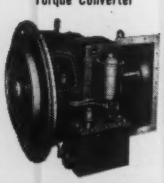


This 3-ton mine locomotive, shown partially assembled at the plant of The Ruth Company, employs a Twin Disc Hydraulic Power Take-off for smooth operation.

Twin Disc Torque Converters, which multiply engine torque up to five times, are used on Allis-Chalmers heavy-duty tractors.



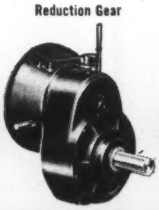
Heavy Duty Clutch



Hydraulic Torque Converter

Tractor Clutch

TWIN DISC
CLUTCHES AND HYDRAULIC DRIVES
REG. U. S. PAT. OFF.



Reduction Gear

Marine Gear

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76. Free-Cutting Steel

Jones & Laughlin Steel Corp.—8-page illustrated revised edition of booklet on E-Steel gives complete description of this Bessemer screw stock. To results of field tests, booklet adds additional material from industry-wide experience with this free-cutting steel.

77. Internal Grinding Spindles

Pope Machinery Corp.—72-page illustrated catalog No. 57 is designed to aid user to make quick accurate selection of correct internal grinding spindle for any application. Data covering large number and variety of spindles are indexed by type of spindle, by make of grinding machine to which spindle applies and by style numbers.

78. Thermosetting Plastics

Continental-Diamond Fibre Co.—30-page illustrated catalog DC-50 presents complete data on Dilecto laminated thermosetting plastic developed to meet needs for waterproof electrical insulating material possessing great mechanical strength and adaptability to all fabricating operations. Also covered is Celoron fabric-base laminated and molded thermosetting phenolic plastic.

79. Shims, Lock Nuts & Stampings

Laminated Shim Co. — 8-page illustrated booklet "Shims, Lock Nuts, Stampings" describes Laminum solid shims that peel for adjustment, An-cor-lox lock nuts using soft nonspringing metal ring at base for locking device, and stampings which can be supplied in any material that can be punched.

80. Beryllium-Copper Products

Beryllium Corp.—Two bulletins Nos. 12 and 13 present instructions on how to order beryllium-copper strip and how to specify rod, bar and wire, respectively. Discussion of available alloys, conditions and tempers is included together with tables covering mill sizes and properties.

81. Steel Shafts

De Laval Steam Turbine Co., Imo-De Laval Products Div.—20-page illustrated bulletin G-7 contains formulas, curves and tables that will asssist designers in determining proportions, stresses, combined torsional and bending loads, etc. in round, square, hollow, keyway and stepped shafts.

82. Aluminum Products

Revere Copper & Brass Inc.—Folder "Properties of Revere Aluminum Alloys" contains condensed but comprehensive chart which gives properties of most of aluminum alloys available in form of wrought products.

83. Control Valves

MacMillin Engineering Corp.—4-page illustrated pamphlet "Electraulic Control Valves" deals with control valves which are hydraulically balanced, both laterally and axially, for effortless control. They can be arranged for any cycle of press operations and can be mounted at any angle wherever desired.

84. V-Belt Drives

Browning Mfg. Co.—32-page illustrated catalog No. V-168-A contains engineering data, horsepower ratings and other information on steel cable Gripbelt V-drives. Tables present hundreds of practical stock drive combinations which have been worked out for convenient selection.

85. Shaded-Pole Motors

Fasco Industries, Inc.—Illustrated leaflet on six-pole shaded-pole motors available in sizes from 1/30 to ⅛-hp presents chart of performance data, list of dimensions and performance curves.

86. Large Disk Clutch

Conway Clutch Co.—8-page illustrated bulletin 15L describes and points out construction details of large size heavy duty clutch available in single, tandem and triple plate designs. Sizes include 14, 16, 18 and 20-in. diameters.

87. Power Distribution System

Trumbull Electric Mfg. Co.—40-page illustrated catalog TEB-1 is descriptive of LVD and SAC heavy-duty Flex-A-Power power distribution systems which are used as main feeders. Comparative analysis between Flex-A-Power and standard wire and conduit systems on basis of cost and performance is made. Covered also are construction details, accessories, engineering data and application data.

88. Air Line Filter

Skinner Purifiers, Inc.—4-page illustrated bulletin No. 631 presents description and instructions on use of model A-1 filter for removing free water, oil and solids from air and gases for protection of precision parts, valves and other instruments.

89. Explosionproof Motors

Reliance Electric & Engineering Co.—Illustrated leaflet C-130 discusses three classes of explosionproof motors with particular emphasis on class I group D types made for application in locations where gasoline, oil, naphtha, alcohols, acetone, lacquer solvent vapors and natural gas are present.

90. Flash Butt Welding

American Welding & Mfg. Co.—20-page illustrated booklet No. CA-50-75 discusses advantages of flash butt welding by controlled techniques and depicts typical product applications. Products and subassemblies covered range in diameter from 4 to 100 in. and in length up to 16 miles.

91. Speed Reducer

Dodge Mfg. Corp.—16-page illustrated bulletin A-470 presents cutaway views, technical data and prices of Torque-Arm speed reducer which is available in six sizes with capacities to cover wide range of drives up to 25 hp. It consists of reducer with fixed ratio of 15:1 driven by motor through V-belt drive.

92. Welding Rods

Eutectic Welding Alloys Corp.—Measuring 2 x 3 ft, welding wall chart lists over 100 EutecRods and EutecTrodes with their standard sizes, giving for each rod such data as: type of joint for which suitable, metal on which used, bonding temperature, tensile strength, Brinell hardness and degree of color match. Check list of 300 production applications is included also.

93. Riveted Roller Chain

Baldwin-Duckworth Div., Chain Belt Co.—8-page illustrated bulletin No. 50-6 describes Baldwin-Rex BA riveted roller chain assembly consisting of standard riveted roller chain with special single-pin connector link. Connector has pin which can be driven out easily to assemble or disassemble chain.

94. Electronic Controls

Photoswitch Inc.—65-page illustrated booklet entitled "Cutting Production Costs with Electronic Controls" contains 45 case studies describing actual examples of cost-saving production techniques involving use of photoelectric or electronic controls.

95. Flexible Shaft Equipment

Stow Mfg. Co.—Illustrated catalog No. 50 lists details and specifications of complete line of flexible shaft machines and accessories and tool, power drive and drill shafts. Machines and shafts are shown by type and use.

FOR MORE INFORMATION

on developments in "New Parts" and "Engineering Department" sections—or if "Helpful Literature" is desired—circle corresponding numbers on either card below

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96. Electrical Contacts

Fansteel Metallurgical Corp.—36-page illustrated booklet M-105 contains information on electrical contacts of value to design engineers. Subjects covered include contact design, materials, properties and principal uses. Assembly methods such as riveting, spinning, upsetting, brazing and spot and projection welding are discussed also.

97. Electric Motors

Brown-Brockmeyer Co. — 48-page catalog "The Complete Line" consists of series of bulletins containing information on repulsion-start induction-run, Dyna-Line brush-lifting repulsion-induction, Power-Poise capacitor-start induction-run, split-phase open ventilated, polyphase squirrel-cage induction, gearhead and other types of motors.

98. Rotary Feed Table

Bellows Co.—8-page illustrated bulletin T-25 gives detailed information on how both mechanically operated and electrically operated rotary feed tables can be used for fast feeding of work pieces to machine tools in order to lower production costs.

99. Locking Pin

Driv-Lok Pin Co.—16-page illustrated catalog "Driv-Lok—The Pin That Locks" describes assembly device that performs functions of taper pin, key, set screw, straight and cotter pin or rivet. It is made from cylindrical metal stock with four parallel grooves impressed longitudinally on exterior surface. Metal displaced forms flute which exerts radial force against hole wall to provide positive self-locking.

100. Stress Analysis

Baldwin Locomotive Works—12-page illustrated bulletin 279-A tells how to select and use SR-4 bonded resistance wire strain gages in stress analysis. It explains fundamentals of both simple and practical strain gage circuits and instruments.

101. Powder Metallurgy

Wel-Met Co.—16-page illustrated booklet No. 104 contains fundamental technical data on powder metallurgy and depicts bearings and parts produced from powder metals. Mechanical and physical properties are covered, and advantages and limitations are discussed.

102. Speed Reducers

Perkins Machine & Gear Co.—2-page illustrated leaflet "Perkins Speed Reducers" describes size 250 worm gear units available with either left or right hand assembly or with double extension shaft. Reduction ratios from 5-1/6:1 to 56:1 are offered.

103. Oil-Hydraulic Pumps

Lear, Inc., Romec Div.—2-page illustrated product data sheet No. 7-10 reviews series RG-7380 oil-hydraulic pumps which deliver 5 gpm at 1800 rpm and operate at 1000 psi continuous duty or 1500 psi intermittent duty. Operating features are described and specifications listed.

104. Air Cylinders

Rivett Lathe & Grinder, Inc.—16-page illustrated catalog section No. 54 is descriptive of entire line of Gerotor air cylinders. Drawings and tables are furnished for each model and size cylinder showing complete detailed dimensions. Types and model valves most suited to various applications are also recommended.

105. Castings & Patterns

Howard Foundries—72-page illustrated brochure "Castings and Patterns by Howard" shows facilities for production of permanent mold, aluminum, magnesium, bronze, semi-steel and electric alloy steel castings. Sections are devoted to each of above types and methods of producing them.

106. Hydraulic Transmission

Vickers, Inc.—2-page illustrated bulletin No. 47-40 features series TR2-R13F13 variable speed hydraulic transmission which features stepless speed change from zero to maximum output. Having automatic overload protection, unit delivers output shaft speed in either direction without changing input shaft rotation. Full torque is available throughout entire speed range.

107. Fractional Horsepower Motors

Westinghouse Electric Corp.—4-page illustrated booklet B-3075-C deals with selection of fractional horsepower motors. Included is chart which presents in tabular form the various types of small motors, together with complete data concerning horsepower range, speed characteristics, torque, reversibility, bearings, mountings and application.

108. Time Cycle Controller

Taylor Instrument Cos.—16-page illustrated bulletin 98154 features Flex-O-Timer time cycle controller which is particularly adaptable to those applications which have multiplicity of functions to perform during course of a cycle. Five models are available for process co-ordination applications requiring all-electrical or all-pneumatic operation, or any specified combinations of each.

109. Bronze Stock

Buckeye Brass & Mfg. Co.—Pocket-size catalog No. 480 lists fully-machined bronze maintenance bars, solid maintenance bars and standard sleeve type bearings in handy reference form. Sections are devoted to size and numerical listings and to special operations requiring drilling and grooving for oil or grease lubrication.

110. Hydraulic Cylinder Control

Hydraulic Equipment Co.—2-page illustrated file TV30-10149 explains Hydreco model TV 30 combination tank and valve unit which employs one to three plungers to control individually single or double acting hydraulic cylinders. Control device has maximum capacity of 14 gpm.

111. Selenium Rectifiers

Accurate Engineering Co.—16-page illustrated bulletin 300 describes series 28 single and three phase selenium rectifiers for use with any equipment requiring direct current. Six types have outputs of 115 or 230 v dc in capacities from 57 to 80 kw. All models incorporate selenium rectifying element, full-wave bridge circuit, ebony panelboard and thermal protection.

112. Capacitor-Start Motors

Emerson Electric Co.—4-page illustrated index B580 covers capacitor-start motors in sizes from ⅛ through ¾-hp. Quiet single-phase units provide high starting torque, constant-speed operation and thermal overload protection. Construction features are depicted and complete range of sizes and mountings available is tabulated.

113. Tractor Overdrives

Hupp Corp. — 4-page illustrated folder "Match Your Tractor to Your Job" is descriptive of Balanced Power tractor overdrive or dual under-overdrive which doubles and triples, respectively, number of available speeds of tractor. Fitting between transmission and clutch, unit affords efficient and economical engine speed.

114. Solenoids

West Coast Electrical Mfg. Co.—4-page illustrated folder "WesCo A-C Solenoids" and enclosed price list cover 200 series which has 4 to 20-lb pull or push and 1-in. plunger travel. Units can be furnished for operation on 24 to 440 v, 25 to 60-cycle current.

115. Shaded-Pole Motors

Redmond Co.—4-page illustrated circular "Redmond MicroMotors" refers to low cost type L lightweight shaded-pole alternating current motors with open ventilation. Usable for continuous duty, they are made in 1/40, 1/70 and 1/150-hp sizes, with special windings available for intermediate ratings.

116. Industrial Plastics

Union Carbide & Carbon Corp., Bakelite Div.—32-page illustrated special edition of "Bakelite in Review" recounts forty years of progress in plastics industry. Examples are cited of role of plastics in development of modern industries and contributions of this company in its field.

117. Electrical Contacts

Gibson Electric Co.—Illustrated circular No. 501 deals with Steelback electrical contacts which consist of silver facing individually bonded to steel backing for projection welding to contact supports.

118. Bronze Bearings

Johnson Bronze Co. — 20-page illustrated manual L-5 is partial listing of stock sizes of Ledaloyl self-lubricating bronze bearings. Powder-metal bearings are of copper, tin, graphite and lead combined in form of pre-alloyed bearing bronze.

119. Oil-Hydraulic Pumps

Denison Engineering Co.—2-page illustrated bulletin P-3 is descriptive of HydrOILic axial piston constant displacement pump designed to deliver 9 gmp oil flow at pressures up to 2500 psi. Hydraulically balanced unit can be flange or foot mounted.

120. Hose Couplers

Snap-Tite, Inc.—3-page illustrated monthly publication "Snap-Tite Bulletin" features one particular series of hose coupler per issue and discusses applications, laboratory test data and engineering phases of each type. Price list and complete specifications are included.

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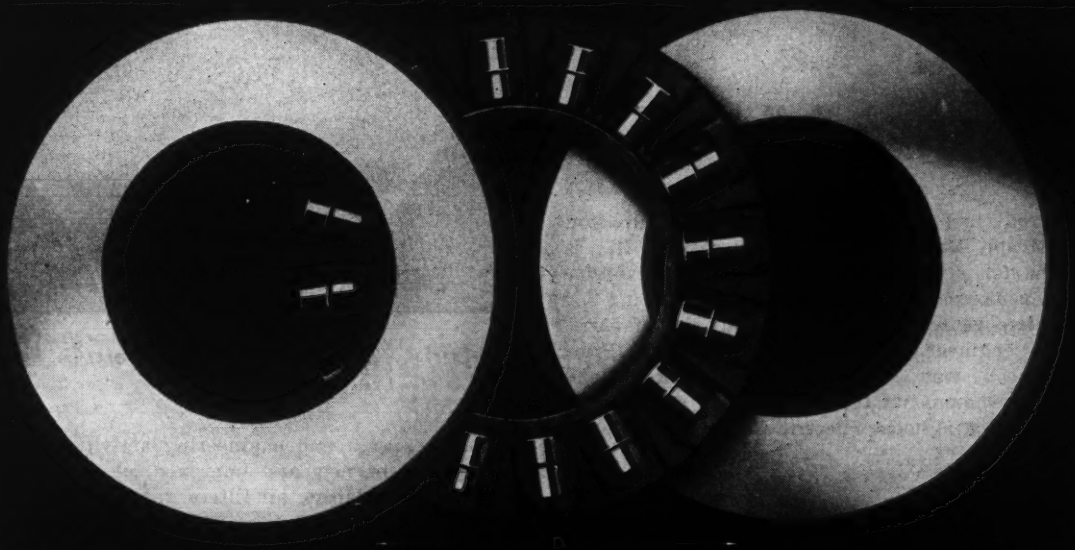
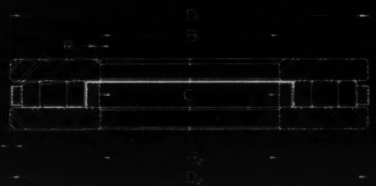
KAYDON TYPE RT THRUST ROLLER BEARINGS



STRAIGHT ROLLER TYPE
FLAT RACES
TYPE RT



SIZES AS LISTED
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Bearing No.	Sizes and Dimensions • See Drawing Above						Roller Diam.	R 45° Chamfer	Lbs. †Capacity at 100 R.P.M.
	B_1	B_2	C	D_1	D_2	H			
RT-26	2.002"	2.015"	2.010"	5.968"	6.000"	1.375"	½"	.062"	36,000
RT-27	2.002	2.015	2.010	6.968	7.000	1.375	½	.062	48,300
RT-28	2.002	2.015	2.010	7.968	8.000	1.375	½	.062	65,400
RT-36	3.002	3.015	3.010	5.968	6.000	1.375	½	.062	29,800
RT-37	3.002	3.015	3.010	6.968	7.000	1.375	½	.062	43,600
RT-38	3.002	3.015	3.010	7.968	8.000	1.375	½	.062	55,600
RT-39	3.002	3.015	3.010	8.968	9.000	1.375	½	.062	67,500
RT-47	4.002	4.015	4.010	6.968	7.000	1.750	⅝	.062	34,700
RT-48	4.002	4.015	4.010	7.968	8.000	1.750	⅝	.062	54,600
RT-49	4.002	4.015	4.010	8.968	9.000	1.750	⅝	.062	69,400
RT-410	4.002	4.015	4.010	9.968	10.000	1.750	⅝	.062	89,300
RT-58	5.002	5.015	5.010	7.968	8.000	1.750	⅝	.062	40,300
RT-59	5.002	5.015	5.010	8.968	9.000	1.750	⅝	.062	63,300
RT-510	5.002	5.015	5.010	9.968	10.000	2.000	¾	.125	83,300
RT-511	5.002	5.015	5.010	10.968	11.000	2.000	¾	.125	111,600
RT-512	5.002	5.015	5.010	11.968	12.000	2.000	¾	.125	136,300
RT-69	6.002	6.015	6.010	8.968	9.000	2.000	¾	.125	48,300
RT-610	6.002	6.015	6.010	9.968	10.000	2.000	¾	.125	76,000
RT-611	6.002	6.015	6.010	10.968	11.000	2.000	¾	.125	100,100
RT-612	6.002	6.015	6.010	11.968	12.000	2.000	¾	.125	124,300
RT-710	7.002	7.015	7.010	9.968	10.000	2.000	¾	.125	62,400
RT-711	7.002	7.015	7.010	10.968	11.000	2.000	¾	.125	85,800
RT-712	7.002	7.015	7.010	11.968	12.000	2.000	¾	.125	109,200

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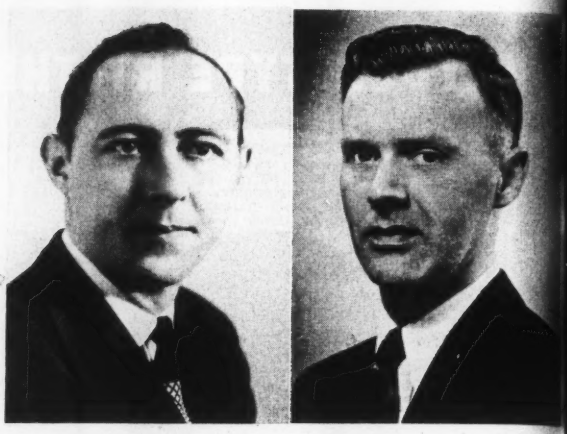


Carl L. Sadler, Jr. Delbert G. Faust

Appointed chief engineer of the hydraulic division of Sundstrand Machine Tool Co., Rockford, Ill., is **Carl L. Sadler, Jr.** A graduate of Purdue University Mr. Sadler has been associated with Sundstrand for the past two years, serving in the capacity of assistant chief engineer of the hydraulic division. Prior to this time he was a group engineer of control and accessory developments for the aviation gas turbine division of Westinghouse Electric Co. in Philadelphia.

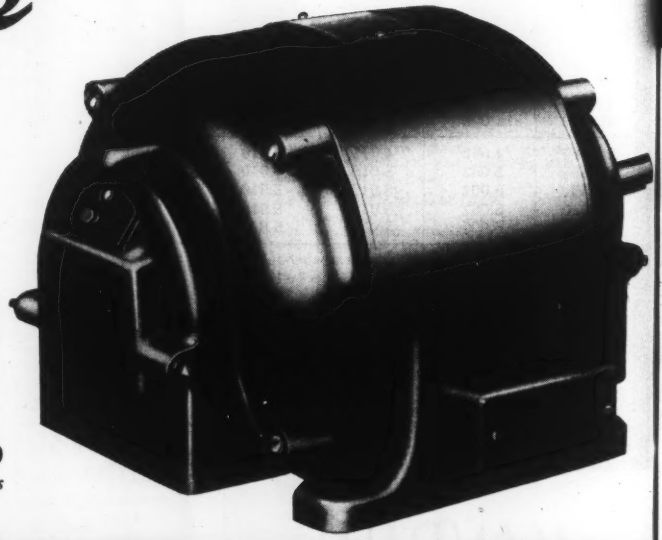
♦

Delbert G. Faust has been appointed chief engineer of the C. A. Norgren Co., Denver manufacturer of pneumatic equipment. Mr. Faust was with Walter F. Kidde Co. for fourteen years, the past ten years of which was spent as chief project and test engineer. For the past several months with the Norgren firm he has been engaged in coordinating engineering operations. In his new capacity he will be in charge of all research and engineering activities in the development of new and improved oil fog lubricators, pressure regulators, air filters, relief valves and other pneumatic products in the Norgren line.

♦

James R. Logan has been appointed vice president in charge of manufacturing and engineering of United States Air Conditioning Corp., Minneapolis, manufacturers of air conditioning, ventilating and heating

HERE'S MOTOR EXPERIENCE THAT CAN BE *yours*



● A high percentage of all Peerless Motors are manufactured to meet the specific operating requirements of the equipment with which they are to be used. Answering these motor problems has given Peerless engineers an experience that can be yours when you call Peerless for advice on motor applications.

You will save time, energy and money by turning your motor problems over to Peerless. Call on us by telegram or telephone. Increased engineering and production facilities can be put to work for you promptly.

THE PEERLESS ELECTRIC COMPANY • WARREN, OHIO

Manufacturers of Quality Motors for more than 55 years

Single Phase • Polyphase • Direct Current Motors

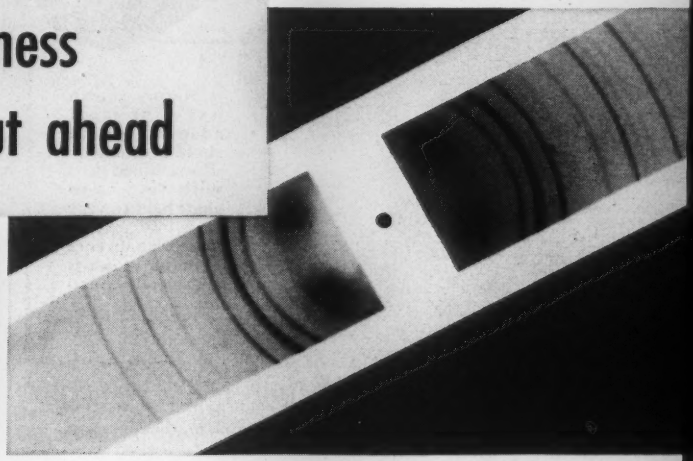
ALL *Peerless Motors* ARE "PEERLESS REGISTERED"

DISPLAYS MASSIVE PRODUCTS—A Diesel locomotive can roar across the Rockies—all on a movie screen in a prospect's office. All because photography can take huge things or small, and make them of a size for a salesman, teacher, or demonstrator to show.

Photography makes big things small—small things big—and business comes out ahead

MAKES MICROSCOPIC DETAILS CLEAR—Photography takes great magnifications produced by the electron microscope (20,000X) on fine-grain Kodak plates, enlarges and records them up to 100,000X on Kodak projection papers. Previously undetectable details and new facts are revealed.

REDUCES FILING SPACE BY 98%—With microfilming, bulky records can be reduced and stored on a few rolls of film. 675 drawings, 24" x 36", can be recorded on a 100' roll of 35mm. Recordak or Kodagraph Micro-File Film. And everything is quickly ready for reference in the Recordak or Kodagraph Film Reader.



REVEALS STRUCTURE AND CONDITION OF METALS—X-ray diffraction patterns on Kodak films or plates provide important information concerning the crystal structure of metals. These patterns help show how alloys can be improved or new alloys made—give data on the effect of machining, drilling, and punching upon the structure of the material.

WITH THE SPEED of a flick of light, photography can reduce or enlarge accurately to scale, and without missing the tiniest detail. And that's not all.

It can magnify time with the high speed motion-picture camera so that the fastest motion can be slowed down for study. It can record the penetrating x-ray and reveal internal conditions of materials and products. With movies and stills, it can repeat a story, time and again, without the loss of a single detail.

Yes, photography serves business and industry in many important and valuable ways. It can work for you, too. If you would like to know how, please feel free to write for literature, or for specific information which could be helpful to you. Eastman Kodak Company, Rochester 4, New York.

FUNCTIONAL PHOTOGRAPHY

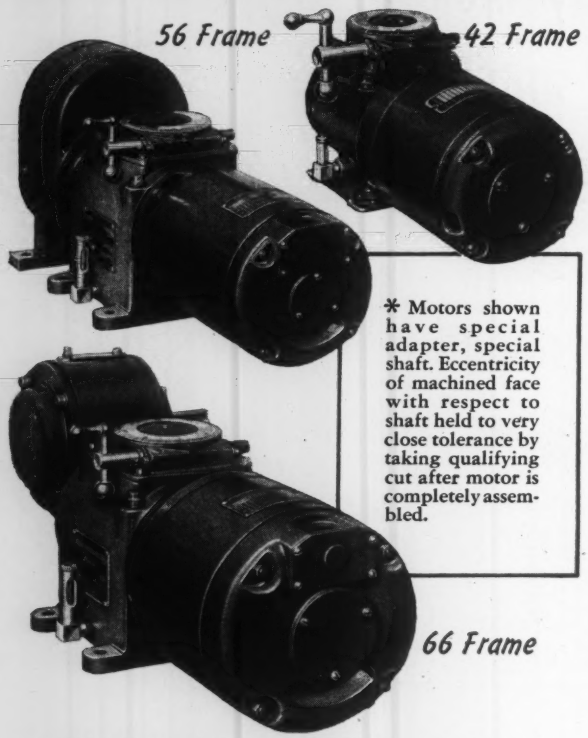
serves industrial, commercial, and scientific progress

Kodak

TRADE-MARK

Now-you can have all three NEMA

FRACTIONAL FRAME SIZES IN STANDARD OR SPECIAL* DESIGNS!



* Motors shown have special adapter, special shaft. Eccentricity of machined face with respect to shaft held to very close tolerance by taking qualifying cut after motor is completely assembled.

GRAHAM TRANSMISSIONS INC. selected this complete line to motorize its widely-used variable speed transmissions. Starting with one frame size only, Graham has been a steady user of Doerr motors for many years.

Many other leading machinery builders continue to place their confidence in Electro Machines products and service as proved by the high percentage of repeat orders.

If you have a particular motor problem please let us hear from you.

NORTH THIRD STREET
CEDARBURG, WISCONSIN

FROM 1/20 to 2 H.P.

DOERR MOTORS

Standard or Designed to Your Specifications

ELECTRO MACHINES, INC.

equipment. At the same time **Merrill L. Benson** was named manager of the coils and coil products division. Mr. Logan has been associated with General Electric Co., General Mills Inc., and Allied Control Valve Co. Inc. since his start in the manufacturing industry in 1929, and Mr. Benson, who has spent more than 27 years with well-known coil producers, is widely recognized as a pioneer in the development of the "plate-fin" coil type.

♦

Appointment of **Charles Mertler** as manager of engineering and production for Stevens Manufacturing Co. Inc., Mansfield, Ohio has been announced. A mechanical engineering graduate of Toledo University, Mr. Mertler has been associated for the past 20 years with Westinghouse Electric Corp.

♦

Joe Wallace is now a design engineer with the Bell Aircraft Corp., Niagara Falls, N. Y. He previously held a similar position with the Curtiss-Wright Corp., Columbus, Ohio.

♦

Wieslaw Jedrzejowski, formerly with H. C. Evans & Co. and Industrial Tool Engineering and Manufacturing Co., is now design and development engineer with J. L. Ferguson Packomatic Co., Joliet, Ill.

♦

J. J. Rozner has been appointed works manager of Aetna Ball and Roller Bearing Co., Chicago. Joining the company's engineering department as a draftsman 21 years ago he has been chief engineer for the past 10 years, a post he will continue to hold together with the supervision of all manufacturing operations.

♦

Appointed vice president of engineering for Packard Motor Car Co., Detroit, is **William H. Graves.** He joined Packard in 1919 and has served as administrative head of the engineering division since last April when he was named vice president and director of engineering.

♦

Dudley Chambers has recently joined the International Harvester Co., Fort Wayne, Ind., as a principal design analyst in the advanced engineering section.

♦

J. E. Ryan and **C. P. Hayes** have been named staff assistants to **B. A. Case,** manager of engineering of General Electric's Small Apparatus Divisions, Schenectady, N. Y. Joining the company in 1934, Mr. Ryan was appointed supervisor of the mechanical design course in 1937 and in 1940 was transferred to

New! HIGH FREQUENCY

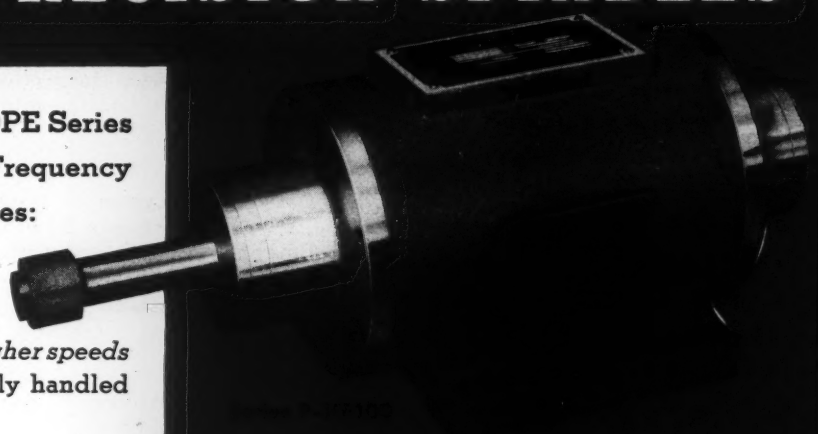
POPE PRECISION SPINDLES

Specify POPE Series P-HF100 High Frequency Motorized Spindles:

—when you need *higher speeds* than can be properly handled by belt drive.

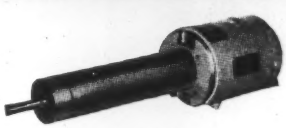
—when *more efficient* internal grinder spindles are wanted.

—when a *more compact* spindle package is desired.



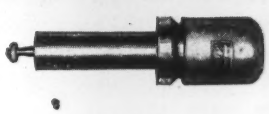
POPE High Frequency, Series P-HF100 Internal Grinder Spindles are regularly furnished in speeds from 5400 to 54,000 R.P.M.

POPE designs and builds *special* High Frequency Spindles for a wide range of applications, a few of which are described below.



P-5681

POPE High Frequency Direct Motorized Cartridge Type Spindle, with 1 HP motor at 7200, 10,800, 14,400 and 21,600 RPM, using 240 and 360 cycle current in combination with a two-pole, four-pole motor.



P-693

High Frequency 16,600 RPM totally enclosed fan cooled 1 HP motorized Spindle, over-all length 18 3/16", barrel dia. 3", barrel length 9".

P-1932

High Frequency 18,000 RPM totally enclosed, fan cooled ½ HP motor Spindle arranged for vertical operation with the grinding wheel or tool at the upper end. Length 15 15/16", barrel dia. 2½", barrel length 7 13/16". Pope sealed-in lubrication.

P-5552

POPE Heavy Duty, High Frequency, Direct Motorized, Cartridge Type Milling Spindle. 14,400 RPM, 15 HP totally enclosed, water cooled motor, 240 cycles, 3 phase. Oil mist lubrication.

When You Want A High Frequency Spindle, Let Us Have Your Specifications And We'll Provide Recommendations And Quotations.

No. 67

POPE POPE MACHINERY CORPORATION ESTABLISHED 1888 501 RIVER STREET • HAVERHILL, MASSACHUSETTS BUILDERS OF PRECISION SPINDLES

A Cambridge One Direction Weave Belt, chain driven and fitted with stiffening rods, carries raw wool through the continuous drying operation. The wide-open mesh permits free passage of the warm, dry air for fast, clean drying.

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WIRE BELTS

Cambridge Wire Belts are efficient workers in your field. They serve in heat, cold, corrosive and wet processes; through packaging and assembly operations—moving quickly or slowly at constant or varying speeds. Wherever a material can be processed during movement, a Cambridge belt can, and does, save time, money and floor space.

Cambridge offers you a modern engineering staff with experience in your field and skilled workmen to produce the right belt in the correct mesh and weave, metal or alloy to perform your operation better . . . for longer.

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OFFICES IN PRINCIPAL INDUSTRIAL CITIES

the control engineering division where he worked on design of control devices until 1945 when he was made head of the division's general research group. He was appointed engineer in charge of general developments for the control division laboratory in 1947 and was named engineer in charge of the laboratory last year. Mr. Hayes joined the company in 1932 and was transferred to the specialty transformer division at Fort Wayne as a design engineer. He went to the fractional horsepower motor engineering division in 1937 where he stayed until 1946, reaching the position of assistant section engineer. He was then made division engineer of the ballast division, a post he held until this appointment.

♦

Lyman C. Fisher, formerly associate chief of the Mechanics Research Division, Research Department, U. S. Naval Ordnance Laboratory, White Oak, Md., has been appointed chief of NOL's Engineering Department II.

♦

William F. Burrows recently joined White Motor Co., Cleveland, as an assistant engine design engineer. He had been assistant chief engineer with the Air-cooled Motors Corp., Syracuse, N. Y.

♦

Now a design engineer with the MB Manufacturing Co., New Haven, Conn., is **Harry N. Cottle, Jr.** Prior to this he was project engineer in the Propeller Division of Curtiss-Wright Corp., Caldwell, N. J.

♦

John W. Tawresey has joined the Barden Corp., Danbury, Conn., as assistant to the president. He was formerly vice president in charge of engineering with the Bunting Brass and Bronze Co., Toledo, Ohio.

♦

Edward C. Yokel is now assistant chief engineer with the Twin Disc Clutch Co., Racine, Wis. He formerly held a similar position with the company's hydraulic division at Rockford, Ill.

♦

Harry R. Seelen has been appointed manager of the Lancaster Engineering Section of RCA Tube Department to succeed **Dayton Ulrey.**

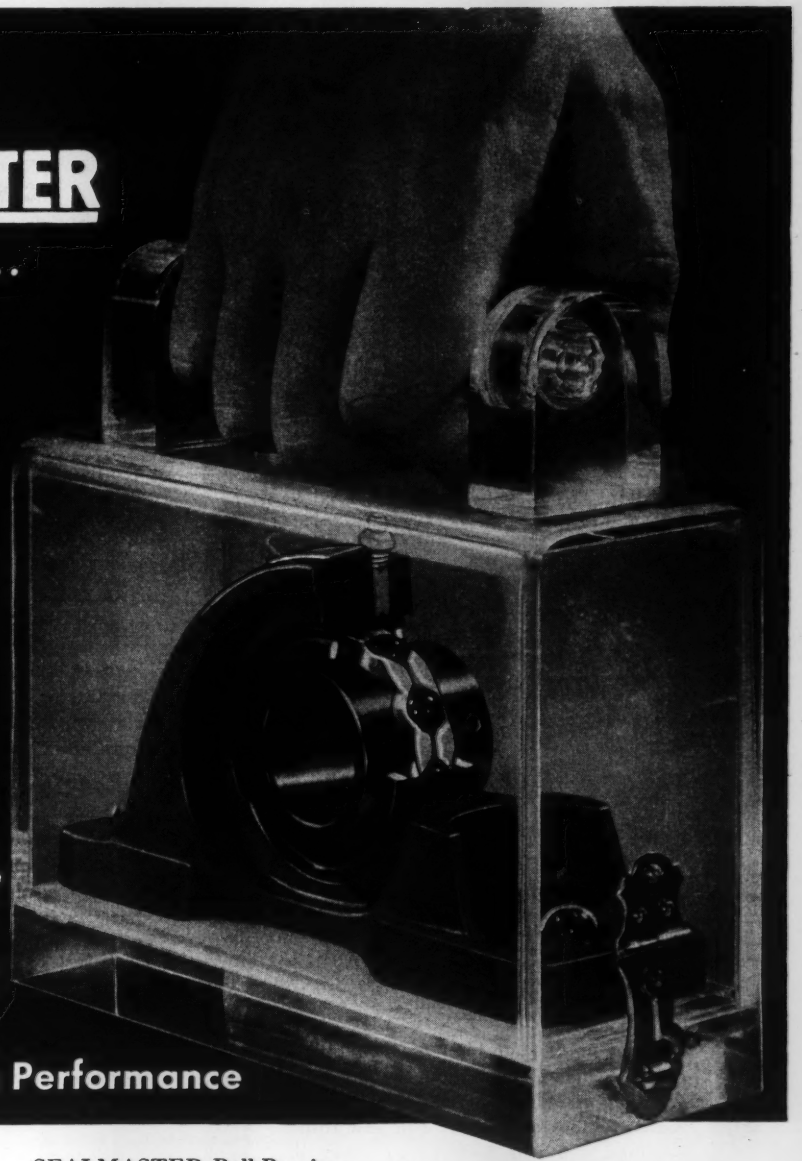
♦

Robert P. Merritt and **Ernest W. Smith, Jr.** were elected president and executive vice president, respectively, of Hartford Special Machinery Co., Hartford, Conn. Mr. Merritt succeeds his father, the late Joseph Merritt, one of the founders of the company. At the same time William H. Storrs was elected vice president in charge of engineering.

Ask the

SEALMASTER

Engineer for this...



Insight

on Improved Bearing Performance

He will show you how . . . and why . . . SEALMASTER Ball-Bearing Units help make machines run longer . . . smoother . . . and with less maintenance. You'll learn why so many leading manufacturers and product designers use and specify SEALMASTER Ball-Bearing Units in their plants and on their products.

So, listen to the SEALMASTER bearing specialist. There is one near you. And if you have a *special* bearing problem, put it up to him for recommendation. SEALMASTERS are made in Pillow Block (shown above), Cartridge, Flange, Flange-Cartridge, Take-Up and Hanger Units as well as many special types. Write us and we'll arrange a quick appointment. Also, there's a catalog available. Write for it today.

5 Reasons Why You're Better Off with SEALMASTERS

1. *Permanently Sealed!* Felt-lined steel flinger rotating in labyrinth, prevents entry of dirt and retains proper amount of lubricant.
2. *Self-Aligning!* Bearing unit, with seals independent of the housing, can align itself in any direction without seal distortion.
3. *Prelubricated!* The bearing chamber is filled with the proper amount of grease before bearing leaves factory.
4. *No Housing Wear!* Patented locking pin and dimple prevent rotation of outer race in housing, thus eliminating housing wear . . . permit shaft alignment and position unit for relubrication.
5. *Floating Retainer!* Ball retainer is designed to float on ground inner surface of outer race . . . traps lubricant and prevents churning.

Bearing Division

STEPHEN S-A DAMSON MFG. CO.

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PRECISION COUNTS
...COUNT ON...

Standard and Special Ball
Thrust Bearings • Angular
Contact Ball Bearings •
Spherical Roller Bearings
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• Hardened and Ground Washers • Bushings

● A survey of your high precision parts requirements by Aetna engineers may bring you important economies. For, here at Aetna, besides top quality ball and roller bearings, you will find today's most advanced facilities for producing an infinite variety of miscellaneous precision parts . . . to whatever hardness or precision finish you need . . . at prices usually below your own production costs. Complete laboratory control and exceptionally strict inspection procedures will guard the quality of your parts from start to finish. Our variety of tools and dies are so extensive that most parts can be produced to fit your requirements without extra expense of special tooling . . . even should you require washer-type parts up to 38" O.D. Send your prints for recommendations and estimates. No obligation.

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NEWS
OF MANUFACTURERS

Representatives of the **Prestole Corp.**, Toledo, O., and the **Westinghouse Corp.**, held a "Fastening Clinic" recently at Westinghouse's Mansfield plant. Primary objective of the clinic was to demonstrate advanced fastening methods and how these methods reduce assembly, materials and labor costs. Prestole fasteners and the "Prestolizing" method of fastening, which eliminates the handling of separate fasteners, were demonstrated. Prestole officials report that they have received requests from other manufacturers for such clinics to benefit their assembly engineers.

♦

The **George K. Garrett Co.** has inaugurated a new expansion program with the addition of a new plant located in Elwood City, Pa. Covering more than 50,000 sq ft, the plant has facilities for producing flat washers of all types, small and medium stampings, and complete assemblies.

♦

Riverside Metal Co., Riverside, N. J., has just completed expansion of its mill facilities, including new casting equipment, annealing furnaces, and materials handling equipment in addition to wire drawing and rolling mill equipment.

♦

Vonnegut Moulder Corp., Indianapolis, Ind., has purchased the patent, sales and manufacturing rights for the Newton Swing Belt Grinder from the **Newton Mfg. Co.**, Temple, Tex.

♦

Advance Electric & Relay Co., manufacturer of relays for general circuit control, electronic, aircraft and marine applications, is now occupying a recently completed, 20,000 sq ft plant located at 2435 N. Naomi St., Burbank, Calif.

♦

The partnership of **Binswanger-Henkin Industries** has changed its firm name to **Metalcraft Mfg. Corp.** and its address to P.O. Box 274, 641-643 Union Ave., Memphis, Tenn. The firm provides engineering and industrial design service and manufactures dies, jigs and fixtures, special tools and machines, models and instruments.

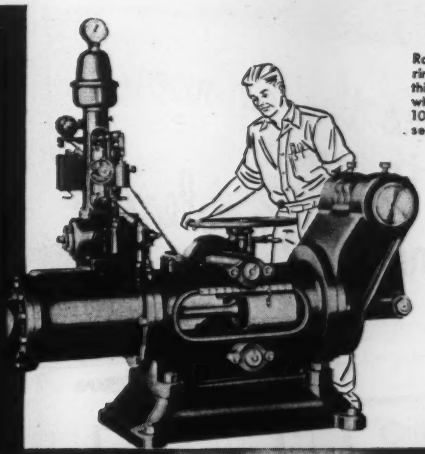
♦

The **Alan Wood Steel Co.**, Conshohocken, Pa., has announced the completion of a modern water treatment plant to purify waste water from the company's steel mills along the Schuylkill river. The installa-

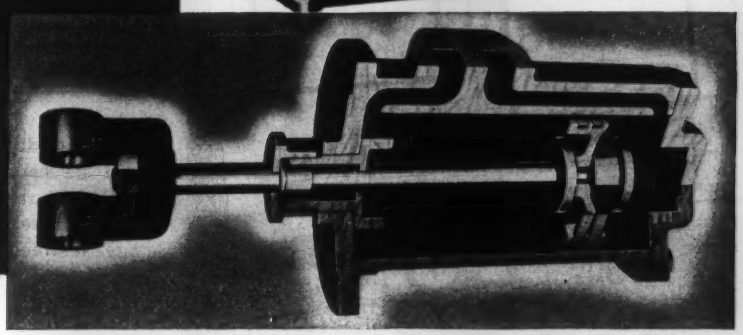
Long Trouble-free Service

with

BURD "Graf-Flox" PISTON RINGS



Rarely, if ever, do piston rings need replacing in this Woodward Waterwheel LHR Governor. Two 10 x 5/8" rings effectively seal the hydraulic cylinder.



Why BURD "Graf-Flox" Rings Offer Profitable Advantages

- Made from fine-grain electric furnace iron, metallurgically controlled—assures best lubricating and wearing qualities.
- Tension cast-in by individual molding method insures long life under extreme operating conditions.
- Precision machined to close tolerances—insures easy accurate fitting, maximum heat dissipation.
- "Graf-Flox" finish—removes "smear" layer from machining operations—provides faster, better break-in surface, eliminates scuffing or seizing.
- Wide range of types and sizes especially engineered and machined to suit your needs.

BURD
Automotive and Industrial
PRODUCTS

Pressure of 200 P. S. I. Sealed Effectively with 2-Ring Piston

The 10" power cylinder assembly of a Woodward Waterwheel Governor, illustrated above, controls generator speeds by regulating water flow. Working pressure of 200 psi is effectively sealed with a two-ring piston equipped with Burd 10" x 5/8" "Graf-Flox" compression rings. The cylinder is bored and reamed to ±.0005" of nominal size. The piston is precision ground .007" under cylinder size. Ring grooves are accurately cut to allow .002" side clearance and .116" depth clearance. The rings are precision turned and are held to ±.001" on width.

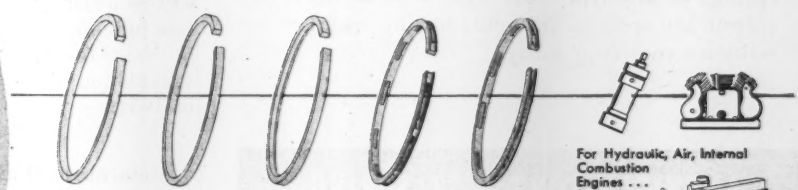
No Run-in Required—Surface Treatment Eliminates Seizing

In addition, the rings are surface treated (Burd's Exclusive "Graf-Flox" Process) which reduces frictional resistance upon installation as much as 60% and eliminates scuffing and seizing. No run-in is required. The "smear" layer resulting from machining operations is completely removed. Natural graphite in the ring metal is exposed to provide a fast sealing outer surface. Cast-in tension and close metallurgical control assure long life after seating. Smooth operating performance and years of trouble-free service are results consistently obtained by Woodward Governor Company.

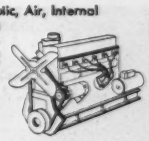
Write For Specification Sheets—Engineering Assistance

Call on Burd engineers for help with your ring design problems. No obligation. Write today for free specification sheets and more facts.

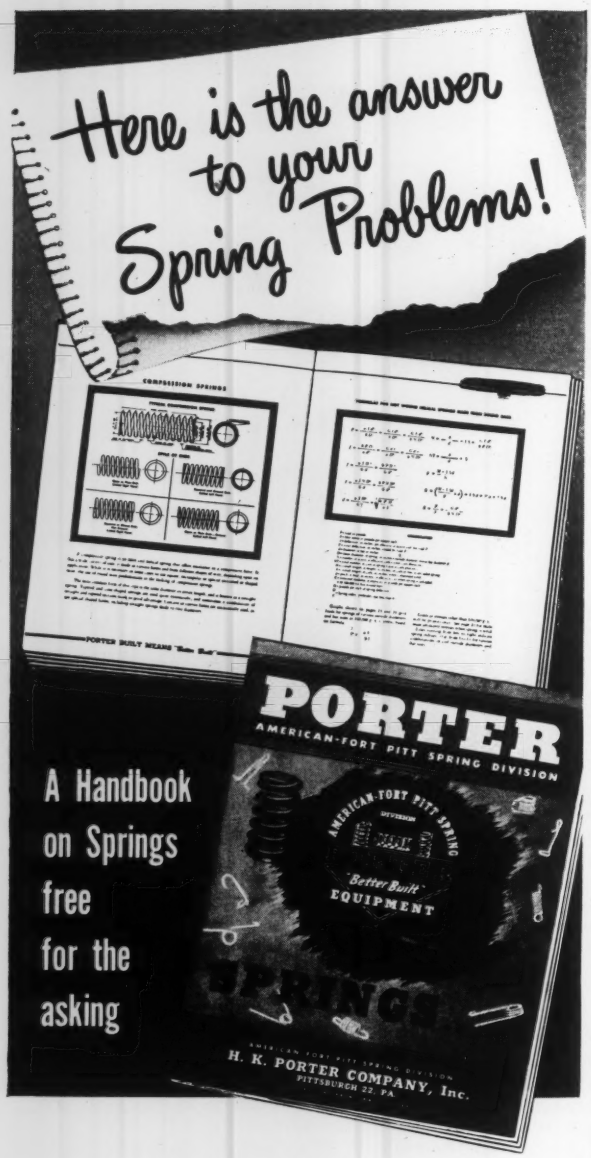
BURD PISTON RING CO. • 23rd Avenue & Gruber Street • Rockford, Illinois



For Hydraulic, Air, Internal Combustion Engines . . . Sizes From 7/8" to 16" Diameter



BUILDERS OF DEPENDABLE PISTON RINGS *Since 1914*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SPRINGS
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If you are designing or making a product requiring a spring in its assembly, you will want this book—a 28-page handbook of engineering data on springs—28 pages of formulas, graphs, charts, tables and drawings. This book will tell you all you need to know about specifying springs of any type from light wire to heavy elliptic leaf springs. It's yours for the asking—write for your copy today.

AMERICAN-FORT PITT SPRING
Division of H. K. Porter Company, Inc.
2 John St., McKees Rocks, Pennsylvania

tion, built to comply with the State's water purification program, not only eliminates the discharge of contaminating waste products from the steel mills, but also acts as a water conservation unit.

•

American Brake Shoe Co., New York, N. Y., has consolidated all operations of its Canadian subsidiaries into one corporation. This new corporation, a wholly-owned subsidiary, is called **Dominion Brake Shoe Co. Ltd.**, with principal office at 1405 Peel St., Montreal, P. Q., Canada. Dominion Brake Shoe will operate all Canadian plants of the parent company through five divisions: Brake Shoe Division, Joliette Steel Division, Canadian Ramapo Division, American Brakelock Division, and Kellogg Division. Products of the new corporation include brake shoes, track equipment, manganese steel castings, automotive brake lining, air compressors and paint spray equipment.

•

A new division for the rolling of magnesium sheet will be established by the **Aluminum Company of America** at its New Kensington, Pa., works in the near future. Increased demand for magnesium sheet has justified resumption of New Kensington rolling operations which were discontinued in 1947.

•

Chief development of the **Panelyte Division of St. Regis Paper Co.**, New York, N. Y., during 1949, was the acquisition of a plant at Kalamazoo, Mich. The plant is being readied for the manufacture of decorative Panelyte (laminated plastics) and molded forms, including refrigerator parts, and should be in substantially full production late in 1950.

•

Westinghouse Electric Corp. has started manufacturing operations at a new plant. Acquired by Westinghouse in March, 1948, as a part of a long-range expansion and improvement program, the plant is located at Hahntown, two miles south of Irwin, and includes 20 acres of land and four 3-story buildings. The property was formerly occupied by the Jersey Cereal Co., division of the General Foods Corp.

•

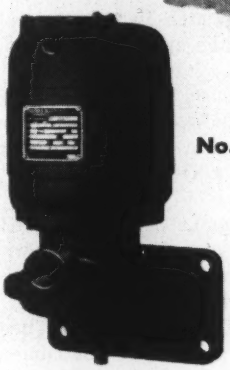
The **Minnesota Mining & Manufacturing Co.**, St. Paul, Minn., has announced plans for the construction of a $3,000,000 office building. The new structure, described achitecturally as functional contemporary, will be adjacent to the present three-level administration building. Each of seven floors, five of which will be built for the present, will contain 34,320 sq ft. Completion of the structure is expected in the summer of 1951.

•

Metalwash Machinery Corp., manufacturer of industrial washing machines, dryers, pickling machines, quench tanks and draw furnaces and special machines, has moved to a new 30,000-sq ft plant located on U. S. Route 1 at North Ave., Elizabeth, N. J.

CENTRIFUGAL PUMPS

that prove their worth continuously



No. 240

No. 206

No. 220

Choice of Styles and Capacities

Brown & Sharpe Centrifugal Pumps are available for vertical and horizontal mounting, for submerged or non-submerged installation, with ¼ and ½ H.P. motors.

Nos. 205, 206, 206A, 207 and 212 are designed for submerged mounting in a tank.

Non-submerged styles include Nos. 220 and 225, and 240 and 245 (for vertical or horizontal installation on tank side).

PERFORMANCE RECORDS of Brown & Sharpe Motor-Driven Centrifugal Pumps come from within our plant as well as from customers. Hundreds of our own machines are equipped with these pumps. Their dependability, durability and adaptability are scrutinized continuously. Actual operating experience confirms the choice of materials, the designs and the construction of these pumps.

Whenever you have a job for a centrifugal pump . . . handling liquids of relatively low viscosities at low heads, for large volumes, or for liquids containing particles of grit and abrasive . . . consider the advantages of pumps that have proved their worth under the most exacting conditions . . . Brown & Sharpe Pumps. Send for latest pump catalog. Brown & Sharpe Mfg. Co., Providence 1, R. I., U.S.A.

IMPORTANT FEATURES

- Simple Construction . . . strong and durable. Balanced hydraulically. Designs provide stability without bearing at the impeller end of shaft.
- Splash-proof motors . . . totally enclosed for protection from vapors, dust and moisture.
- Permanently-sealed ball bearings . . . no lubrication, adjustment, inspection, cleaning or other maintenance needed. Precision seals retain grease . . . lock out dirt and moisture.
- Amply powered . . . sufficient H.P. to handle liquids within specified ranges.

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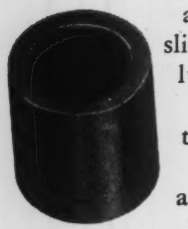
Purebon

solves many problems

caused by sliding or rotating parts which are difficult or impossible to lubricate

A constantly increasing stream of problems are facing engineers and designers today involving sliding or rotating parts where lubrication is difficult or impossible. For such applications, Purebon, the mechanical carbon, is often the ideal answer. Typical applications are seal rings, bearings, pistons, piston rings, pump vanes, valve seats, meter discs, and a host of similar items. Purebon comes in a wide variety of grades. It is strong, tough, readily machineable and in many cases can be molded directly to size.



Bulletin No. 482 tells the complete story of Purebon. Write for your copy today.

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THE ENGINEER'S LIBRARY

Applied Experimental Psychology
Human Factors in Engineering Design

By Alphonse Chapanis, assistant professor of psychology, Wendell R. Garner, director, psychological laboratory, Institute for Cooperative Research, and Clifford T. Morgan, professor of psychology—all at the Johns Hopkins University; published by John Wiley & Sons Inc., New York; 434 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$4.50 postpaid.*

A factor too often neglected in the design of machines is the physical make-up of the human beings who operate them. This book, of pertinent value to the designer and engineer, tells what experimental psychologists have learned pertaining to how men see, hear and move and shows how this information can be applied to design better machines for human use.

To illustrate the treatment of subject matter, the factor 'how men see' is discussed as follows: First the information about man's capacities are presented —what he sees, problems of visibility, visual acuity, eye movements and optical illusions. These data are then applied to the design of instrument dials answering the fundamental questions of good design —what are dials for, do the dials function as intended, what dial shape and dial scale best suit the observer, and what arrangement is most satisfactory? This method of attack is used in essence to illustrate the design of equipment involving sound—radio, telephone, room acoustics, etc.—and of production or processing equipment where motion is studied to effect more efficient operation. Concluding chapters discuss fatigue as induced by operating conditions, work environment, and atmospheric conditions.

To the designer this book offers a wealth of information, a working knowledge of which should help materially in attaining more effective design.

□ □ □

Weld Design

By Harry D. Churchill, professor of engineering mechanics, Case Institute of Technology, and John B. Austin, welding engineer, Republic Structural Iron Works; published by Prentice-Hall Inc., New York; 216 pages, 5½ by 8½ inches, clothbound; available through MACHINE DESIGN, *$6.65 postpaid.*

Today, the process of fabrication by arc welding is being applied to many different types of machine. In order that the designer may be assisted in making a selection on the basis of sound engineering and economy, the authors have correlated into ready reference, information pertinent to weldment design.

Comprehensively, the book treats the analysis of construction materials suitable for machine bases,

From wave to waltz...

via

R/M

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Friction materials in the radio industry? Of course! A sintered metal, specially developed by R/M, is now used in radio controls to speed the selection of wave lengths. In addition . . . as in so many other industries . . . R/M materials are widely used by radio manufacturers in the machining, storage, and transportation of their products. For practical solutions to new problems in brake linings and clutch facings, look to R/M . . . world's largest producer of friction materials!

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Another Contribution to Industry through Powder Metallurgy by BOUND BROOK

IF YOU USE *TINY* PRECISION BEARINGS IN *LARGE* QUANTITIES...TRY "COMPO" SELF-LUBRICATING BEARINGS FOR *CLOSE TOLERANCES AT LOW COST!*

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For products such as electric shavers, clocks, meters and many types of instruments, Self-Lubricating Bearings provide these advantages:

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develops basic relationships to be used in solving the stress complications imposed by welding, and presents informative, well-illustrated discussions on designing arc-welded units, welding specifications, weldery procedure, welding processes, flame cutting and flame hardening, forming of metals, and estimating weldment costs.

An appendix includes tables and data relative to allowable loading on various weldments, minimum weld fillet sizes for different plate thicknesses, and general design analysis. The bibliography provides a list of information sources which can be used effectively in design studies.

□ □ □

Machinery's Handbook

Edited by Erik Oberg and F. D. Jones; published by The Industrial Press, New York; 1911 *pages, 4¾ by 7 inches, clothbound; available through* MACHINE DESIGN, $7.00 *postpaid.*

This fourteenth edition contains a volume of data, practical and theoretical, of permanent value to the designer and builder of mechanical machines and components. As in previous editions introductory divisions present the strength of materials and mechanics of mechanisms with mathematical tables for problem solution. Detailed design discussions of specific machine parts and subassemblies follow. Essentially, all components of drives and transmissions are treated—bearings, belts, cams, clutches, couplings, gears, motors, pulleys, etc. So that design of these components may be adaptable to economical manufacture, production processes, machining standards, small tools, steels and alloys, and inspection devices are discussed

Outstanding revisions of this edition include: Unified and American Standard Screw Thread System, revised to provide interchangeable screw threads in the United States, Canada and the United Kingdom; NEMA motor standards; and chain and sprocket standards.

Manufacturer and Association Publications

Experimental Stress Analysis, Vol. VII No. 1: A group of papers prepared by specialists in the field of stress analysis dealing with stress-strain testing, analysis and evaluation of data derived and methods and apparatus employed. Papers presented are: "Some Experiments on the Bursting of Spherical Rotors by Centrifugal Forces"—J. W. Beams; "Electric Resistance Changes of Fine Wires During Elastic and Plastic Strains"—E. W. Kammer and T. E. Pardue; "BL-310 Strain Analyzer"—Dean Christian; "Tests of a Railroad Bridge with a Mechanical Oscillator"—E. J. Ruble; "Observations and Instrumentation for a Load Test on a Reinforced Concrete Structure"—J. Neils Thompson; "Experimental Determination of Metal Drawing and Forming Forces"—Arthur R. Kimbell; "Mechanics of Skull Fracture"—H. R. Lissner, E. S. Gurdjian, MD, and J. E. Webster, MD; "A Biaxial Stress Machine for the Determination of

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THIS 3¼ INCH DIAMETER CIRCLE SHOWS ACTUAL SIZE CROSS SECTION OF THE 3" BORE O-M CYLINDER . . . .

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O-M CYLINDERS SAVE SPACE PLUS MAINTENANCE AND ORIGINAL COST

• O-M hydraulic, water, air cylinders save up to ⅓ in space required because tie rods and bulky end caps are eliminated. Unique circumferential key locks honed steel tubing to compact end caps. That means *more* power in the space required by conventional cylinders . . . or the same power in less space.

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Plastic Stress-Strain Relations"—Joseph Marin; "Dynamic Measurements During Aircraft Landings"—E. H. Kramer and E. J. Lunney; "Fatigue Life of Rail Webs in Service"—C. J. Code and A. E. F. Billstein; "Superposition in Complex Straining"—K. H. Swainger.

Each of the informative treatments is replete with tabular and graphical data pertinent to the designer and stress analyst. Copies of the 131-page 8½ by 11-in. book may be obtained from the Society for Experimental Stress Analysis, Central Square Station, P. O. Box 168, Cambridge 39, Mass., for $6.00 each.

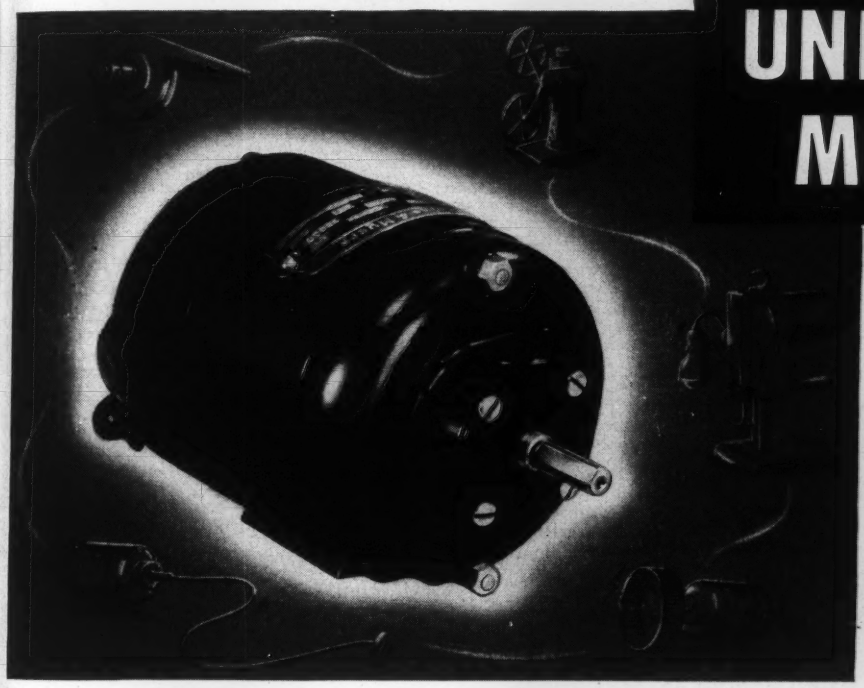
U. S. Industrial Design 1949-1950: Compiled and edited by the Society of Industrial Designers, this 176-page, 8¾ by 12-inch book presents results of styling investigations. The designs are divided into seven categories: Domestic, transportation, commercial, industrial, and personal equipment; commercial interiors and exteriors; and packaging. Sound engineering as well as esthetic principles are considered in the restyling of such equipment as ranges, air conditioners, heaters, cameras, motors, mimeograph duplicators, scales, lathes and chucks, bottle filling and capping machines, tractors, drills, automobiles, trains, ships, etc. Copies may be obtained from the Society of Industrial Designers, 55 W. 42nd St., New York 18, N. Y. Price per copy, $10.00.

Designing with Aluminum Extrusions: A 138-page book which presents excellently the applications of aluminum extrusions in product design. Informatively covered are the manufacturing possibilities, shape design from the manufacturing standpoint, structural design, fits and assemblies of mating extruded shapes, joint design, dimensional tolerances, fabricating properties, finishes, and characteristics of aluminum and the choice of alloy and temper. The text is supported by 16 tables of data on chemical, mechanical and physical properties of aluminum and extrusion process data such as minimum thickness, maximum lengths, etc. Copies may be obtained by addressing company-letterhead requests to the Reynolds Metal Co., 2500 South Third Street, Louisville 1, Ky.

New Standards

A new 12-page American Standard, *Gaging Practices for Ball and Roller Bearings,* B3.4-1950, has been approved. In general, the Standard specifies the most acceptable methods of determining whether bearings conform to specified dimensions. Specifically, the Standard defines inspection methods, giving gaging loads and methods of measurement. Standard methods are given for obtaining dimensions of inner and outer rings, parallelism of sides, eccentricity, groove parallelism with sides, and other significant measurements. It also prescribes methods of measuring internal fit and tolerances for side run-out of lock nuts. Copies of the American Standard may be obtained from the American Standards Association, 70 E. 45 St., New York 17, N. Y., at 50 cents per copy.

YOU'LL LIKE THESE R & M UNIVERSAL MOTORS



They Help You Solve Your Problems!

High starting torque; high operating speed; adaptability to speed control—to *these* inherent characteristics of series motors Robbins & Myers has added *many performance advantages.* New, completely redesigned R & M universal motors, from 1/100 to 1/3 horsepower, are ready *now* for powering your small equipment.

BETTER, INSIDE AND OUT

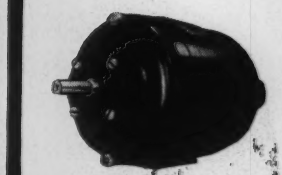
From their *dynamically balanced* armatures to their rigid, welded-steel shells, nothing has been slighted. Commutation, brushes, bearings, insulation, and ventilation all represent *true progress*—progress backed by the competence born of long, successful experience.

STANDARD TYPES—AND "SPECIALS," TOO

In the four frame sizes, there's the *right motor* for every need—sleeve or ball-bearing; fixed rotation or reversible; open or totally enclosed; standard or special ratings . . . and a *welcome variety* of housing types and end-heads.

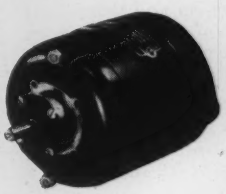
YOURS FOR THE ASKING

We'll be glad to send you literature on these new motors. And, if you would like to have R & M sit in on a discussion of *powering improvements*—that's yours for the asking, too. Address Robbins & Myers, Inc., Motor Div., Dept. E-40. Springfield 99, Ohio.

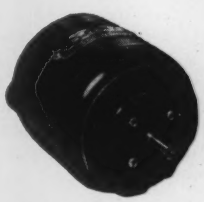


STANDARD PAD BASE

STANDARD LOW BASE
(Also "Medium" and "High" Types)



HUB MOUNTING



FACE MOUNTING

FLANGE MOUNTING

FOR MODERN, COMPACT PRODUCT STYLING—USE

MOTORS by ROBBINS & MYERS

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MOTORS
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Capture Sound Perfectly

REVERE Camera Company, maker of famous REVERE 8mm and 16mm movie equipment, features tone perfection and important mechanical advantages in the REVERE Magnetic Tape Recorder. A high quality FASCO motor assures *quiet* operation . . . eliminates sound distortions caused by variation in speed. Tape starts and stops quickly . . . no reel coasting, no loss or slur of sound . . . fast rewind and forward speeds save time. All *real* advantages . . . examples of efficient, practical FASCO motor performance.

There's a complete line of FASCO Fractional H.P. shaded pole motors from 1/500 to ⅛ H.P. to help *you* improve your product, or pack maximum efficiency and economy into the design of your new one.

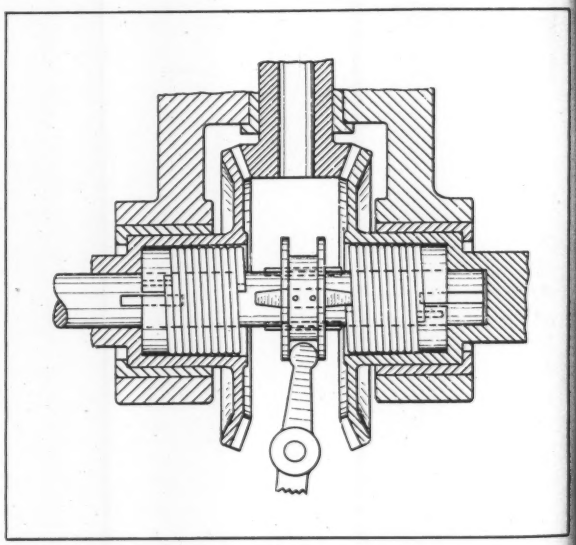
"Your comparison test, too, will prove FASCO best"

WRITE for complete information on FASCO Fractional H.P. Motors.

FASCO
INDUSTRIES, INC.
ROCHESTER 2, NEW YORK

NOTEWORTHY PATENTS

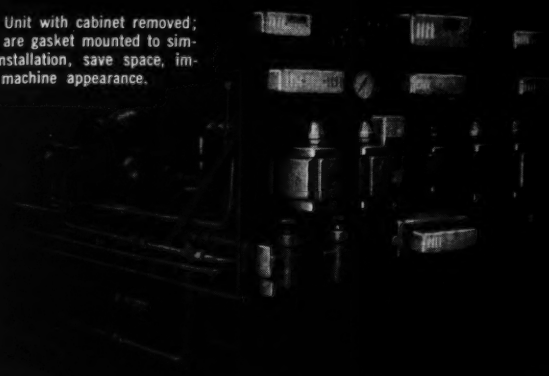
ADJUSTABLE SPRING RELEASE employed in a load limiting clutch can be set to control the maximum torque transmitted by a reversible drive. Drive is from a pinion to two bevel gears and then through either one of two spring clutches to the driven shaft, depending on the desired direction of rotation. Mounted within the hub of each bevel gear and keyed to the driven shaft is a split ring or spring band. One end of a coil spring is pinned to this band. By shifting a sliding collar along the shaft the other end of the coil spring is restrained by a tapered or wedge-shaped projection, causing the spring to expand and contact the inner surface of the bevel gear hub, as in a conventional spring clutch. However, when the torque reaches a predetermined value, the split ring or band opens slightly, winding the coil spring to a smaller diameter. This disengages the spring and



bevel gear until the torque is reduced to a point where the split ring will resume its normal position—at which time the drive is re-established.

The clutch, described in patent 2,487,280 granted to William C. Starkey and assigned to Fletcher Trust Co., can be adjusted to release at a desired torque value. The farther the shift collar is moved from its neutral position the more the tapered projection unwinds the spring, so that a greater torque is required before the split ring will open to release the load.

Smooth clutching and declutching at the release torque are made possible by tapering the hub bore so that the smallest bore diameter is nearest the split ring end of the coil spring. Engagement and release



Power Unit with cabinet removed; valves are gasket mounted to simplify installation, save space, improve machine appearance.

Power Unit having pump driven by standby gasoline engine.

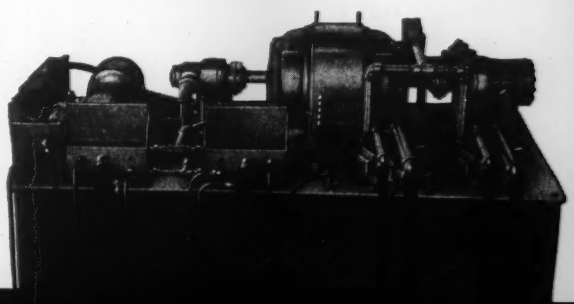
VICKERS Hydraulic POWER UNITS *Custom Built*

Illustrated here are representative examples of the infinite variety of self-contained hydraulic power Units that are designed and built by Vickers for the more efficient operation of many kinds of machinery. These compact Units include all necessary pumps, valves, intermediate piping, reservoir, hydraulic accessories, motors and controls. Hydraulic connections to the machine are grouped in a convenient manifold.

● Design is simplified because arrangement is not limited by structure of machine. Installation time and cost are substantially reduced because all hydraulic equipment is received as a self-contained "package" instead of separate parts that must be individually installed. Every Unit is pretested at factory and is ready for immediate operation. Cabinets which are optional, blend well with modern machine design. Operating adjustments and maintenance are simplified by convenient layout and gasket mounted valves. Vickers Custom Built Power Units have other important advantages . . . ask for Bulletins 47-45 and 46-43.

VISIT OUR DISPLAY AT THE ASTE SHOW — PHILADELPHIA

- **Simplify Design**
- **Reduce Installation Costs**
- **Improve Appearance**
- **Save Time**
- **Easier Maintenance**



Power Unit is placed in base of machine . . . console control where convenient to operator.

VICKERS
Incorporated
DIVISION OF THE SPERRY CORP.

ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921

Application Engineering Offices: ATLANTA • CHICAGO • CINCINNATI CLEVELAND • DETROIT • HOUSTON • LOS ANGELES (Metropolitan) MILWAUKEE • NEW YORK (Metropolitan) • PHILADELPHIA • PITTSBURGH • ROCHESTER • ROCKFORD • ST. LOUIS • SEATTLE • TULSA WASHINGTON • WORCESTER

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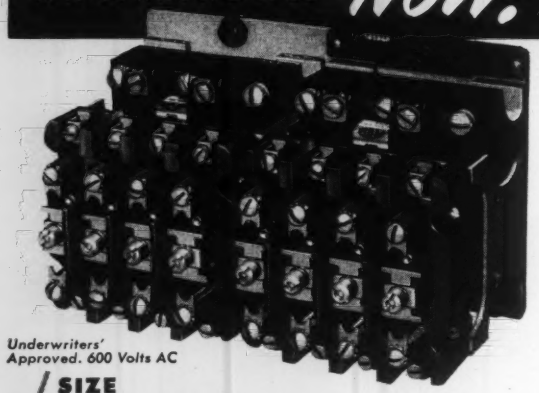


Underwriters' Approved. 600 Volts AC

SIZE

Non-Reversing

2 to 4 Pole 2-3/4" w. x 3-5/8" h. x 3-5/16" d.
5 to 8 Pole 5-9/16" w. x 3-5/8" h. x 3-5/16" d.

Reversing

2 to 4 Pole 5-9/16" w. x 3-5/8" h. x 3-5/16" d.

Note: 10 and 15 ampere contactors have same mounting and overall dimensions.

ACCESSIBILITY

To replace contacts, it is not necessary to disassemble the complete contactor. Just remove the parts comprising the stationary and movable contacts. Contacts can be replaced without disturbing wiring. To change coil, remove magnet frame and coil assembly only. (See illustration below.)

FLEXIBILITY

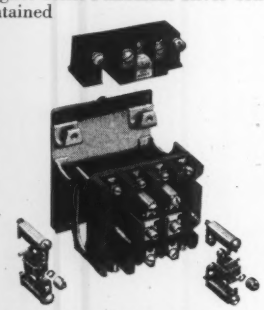
Using a screw driver only, you can easily change any pole from normally open to normally closed. No special parts required. 10 and 15 ampere parts are interchangeable.

RELIABILITY

Laboratory tests involving millions of operations, plus field service of thousands of R-B-M contactors on door operators, radio transmitters, packaging and weighing machinery, hoists, machine tools and many other industrial applications offer proof of dependable, trouble-free performance.

ADVANCED DESIGN

Melamine Insulation. Molded coil housing. Ilsco solderless connectors. 50/60 cycle magnet coils. Palladium silver contacts. Stainless steel self-contained contact springs.



Where space is a factor, and accessibility a must—use R-B-M industrial contactors. Initial low cost plus dependable performance will save you money. Write for Bulletin 600 and price list on your company letterhead.

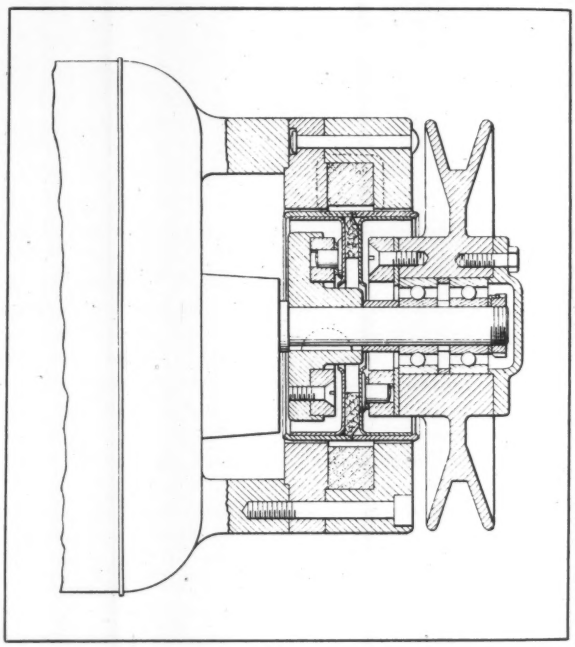
Dept. B-4, R-B-M DIVISION OF ESSEX WIRE CORP.
Logansport, Indiana

RBM

MANUAL AND MAGNETIC ELECTRIC CONTROLS
—FOR AUTOMOTIVE INDUSTRIAL COMMUNICATION AND ELECTRONIC USE

action are initiated at the load-carrying end of the coil spring where clearance between hub and spring is smallest, which eliminates sudden grabbing and chattering of the unit.

CONSTANT AIR GAP between pole faces of a magnetic clutch, regardless of wear of the clutch parts, is provided in an arrangement in which the magnetic coil does not rotate. This stationary coil is surrounded by a U-shaped ring which forms three sides of a flux path. The fourth side is formed by two thin-walled magnetic sleeves, spaced about 0.01-inch from the stationary pole faces and held in axial contact with each other by a disk spring. The magnetic sleeves are, in turn, carried by two friction drums secured to the drive shaft and driven member, respectively. When the coil is not energized, the magnetic sleeves and friction clutch faces will slip relative to one another.



Energizing the coil sets up a flux path through the magnetic sleeves, holding the clutch faces together in driving engagement. Since the sleeves are in continuous rubbing contact, even when the clutch is de-energized, there is never an axial air gap in the magnetic flux circuit. Also, axial wear of the clutch faces does not change the size of the radial air gap between stationary and moving magnetic poles. The patent, No. 2,488,552, has been granted to John G. Oetzel and assigned to Warner Electric Brake Manufacturing Co.

Complete printed copies of all patents are available from the Commissioner of Patents, Washington 25, D. C., for 25 cents per copy.

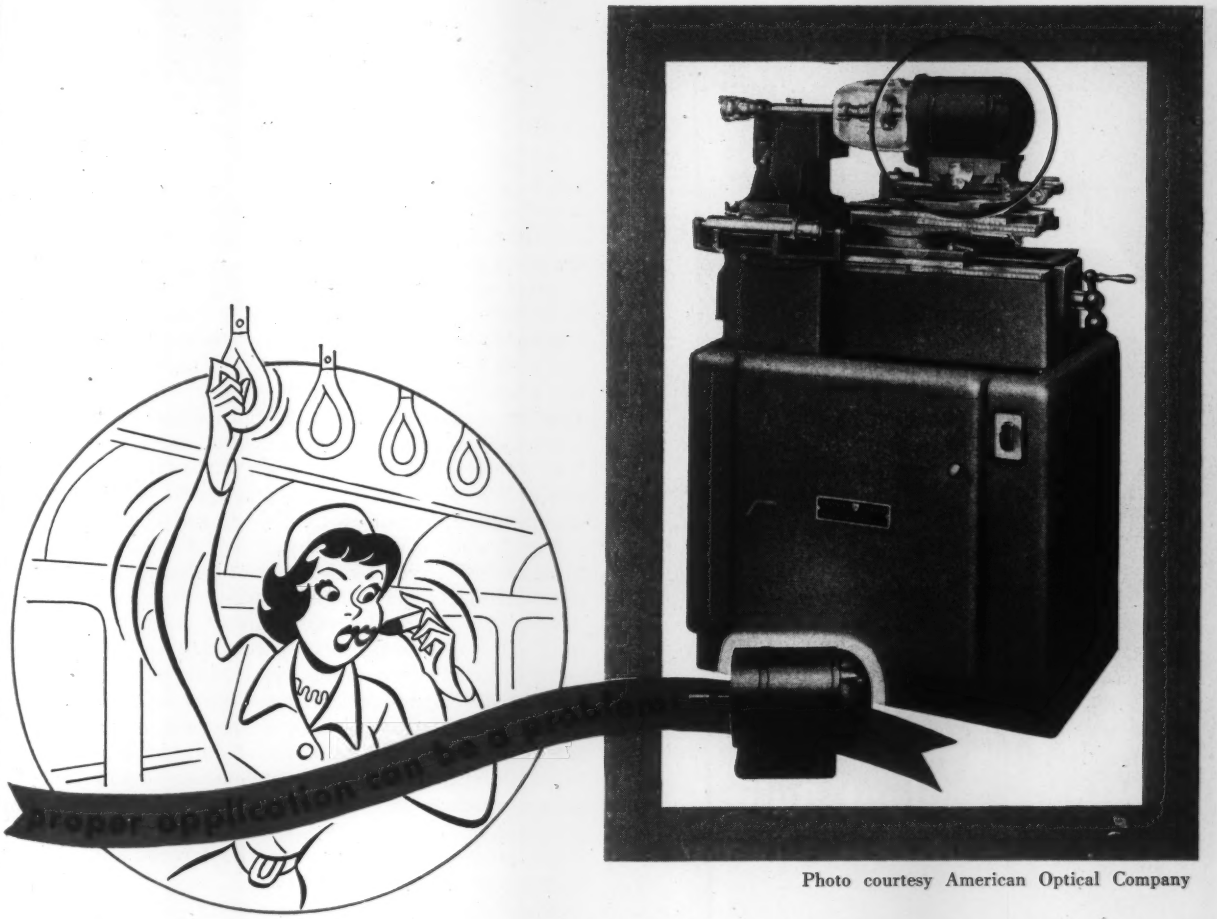


Photo courtesy American Optical Company

. . . take, for instance, the engineering of a suitable small motor for American Optical Company's M405 Universal Micro-Surfacer. This revolutionary optical machine, designed to smooth-grind the concave surface of lenses, demanded most exacting specifications for the motor unit to be used in driving the diamond lap spindle.

- The motor must be totally enclosed, to exclude dust and grinding compounds.
- The motor must be adaptable to a machine bed, similar to a lathe bed.
- Tolerances on center line to base dimension and all parallel tracking surfaces must be held to an absolute minimum, for purposes of interchangeability.
- Parallelisms must be held to an absolute minimum to insure a true running spindle.
- Spindle must have absolute minimum of endplay and runout.

After thorough analysis and study of the problem, Holtzer-Cabot engineers designed a precision motor that not only met all customer requirements, but *was priced competitively*.

This Holtzer-Cabot motorized unit, supplied in various frequencies and voltages as well as in Direct Current, is now being used extensively.

Our files are filled with success stories like this. The application solution described here is just one of many examples of Holtzer-Cabot's versatility and ability to meet customers' most rigid specifications. Holtzer-Cabot motors range from 1/2000 up to 1 H. P.; from 24,000 RPM to 1 revolution per day!

HOLTZER-CABOT

DIVISION OF NATIONAL PNEUMATIC COMPANY INC.

BOSTON 19, MASSACHUSETTS

OFFICES IN NEW YORK, PHILADELPHIA and CHICAGO

GOT AN APPLICATION PROBLEM?

If *you* have an application problem involving motors up to 1 H.P., turn to Holtzer-Cabot for the solution. We are eager to tackle the most difficult motor problem. Write for complete information.

"builders of fine electric motors for three-quarters of a century

semi-knife-edge bearing reduces frictional wear

This is Ward Leonard's new Bulletin 110 Midget Relay for long, trouble-free service, particularly in equipments subject to vibration.

Exceptionally good vibration characteristics are due to proper proportioning of contact masses and springing combined with heavy pressures on both normally open and normally closed contacts.

Higher contact ratings than most midgets. Available up to 3-pole, double throw. Contact finger leads are insulated with the new, impregnated glass-fiber tubing.

Write for Bulletin 110. Ward Leonard Electric Co., 58 South Street, Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

WARD LEONARD ELECTRIC COMPANY

Result-Engineered Controls

RESISTORS • RHEOSTATS • RELAYS • CONTROL DEVICES

Synchronized Drives

(*Continued from Page* 146)

LS_2 is closed by the longer cam and the bridge travel is not interrupted. With the bridge out of alignment, LS_2 does not close the circuit when LS_1 is open and the drive will stop automatically.

After stopping due to dislocation, the bridge must be relocated in the proper position before a new start can be made. To facilitate this relocation, a manually operated selector switch is recommended which opens the primary and the secondary of one of the motors, thus permitting a realignment both ways by the drum-controller and the remaining motor. In its open position, the selector switch shorts all the safety devices except the overload relays in order to make the realignment operation with one motor possible.

Fig. 7 shows a coal loading bridge. Both legs are individually driven by selsyn motors without any mechanical interconnection between the two travel drives. The arrow indicates one of the cams for the automatic bridge alignment control.

To protect the individual motors from overloading, coils of overload relays *OL* are inserted in two of the motor primaries. The coil of the resistor contactor is controlled by the contacts of the overload relays in series with the other safety devices, the differential selsyn contacts and the dislocation limit switches. In case one or both motors are overloaded or one of the other safety devices responds, the drive is stopped by opening the resistor circuit.

The following planning procedure for a two-motor drive is recommended:

1. Determine the maximum torque required for each drive
2. Select the motor according to the maximum torque of the heavier loaded drive and provide two identical motors
3. Determine the maximum and minimum torque difference of both drives occurring during any phase of operation
4. Compare this torque difference with the available margin in *Fig.* 5 with the optimum resistance $R = 0.23\ R_b$ inserted. If the torque difference is lower than the available margin, then this method can be considered adequate; if the torque difference is higher, a selsyn or synchrotie system should be used
5. In laying out the gearing and power transmission, it should be remembered that the actual operating speed will be less than the full motor speed due to the operating resistor approximately $0.23\ R_b$. *Fig.* 5 gives an indication of the speed to be expected, slip is approximately 17 per cent with two integral-horsepower motors fully loaded
6. If a reduction of the inrush current is desired, a secondary resistor up to $0.5\ R_b$ can be used for starting. After coming to speed, the resistance should be changed to $0.23\ R_b$ by the starting device.

The synchronized drive of two wound-rotor induction motors is conveniently simple and inexpensive

and might be convenient for many applications. Operation will be satisfactory under the following conditions:

1. An appropriate rotor resistor, preferably the optimum, approximately 0.23 R_b per phase, should be used
2. The load differences and the minimum and the maximum load should not exceed the limits given in *Fig.* 5
3. Regular stopping should be done by opening the common secondary resistor circuit
4. In cases where rotor displacement is critical, a displacement control by a differential selsyn is recommended
5. In case of a bridge travel, a visual check by the operator, or an automatic alignment control by limit switches, should be provided.

The system is not applicable if drop of speed, poor speed regulation, and reduced efficiency caused by the insertion of the rotor resistor are objectionable.

The system offers sufficient advantages in simplicity and reduced initial cost to justify further study. Interesting items for further study are: Theoretical analysis of the simplified selsyn, usable for determination of the optimum resistor and for prediction of operating characteristics; the possibility of operating two motors of different ratings in a synchronized system and the limitations of such a system; the feasibility of synchronized operation of three or more machines and the requirement limitations for satisfactory operation; and the substitution of reactance for the rotor resistor to reduce losses.

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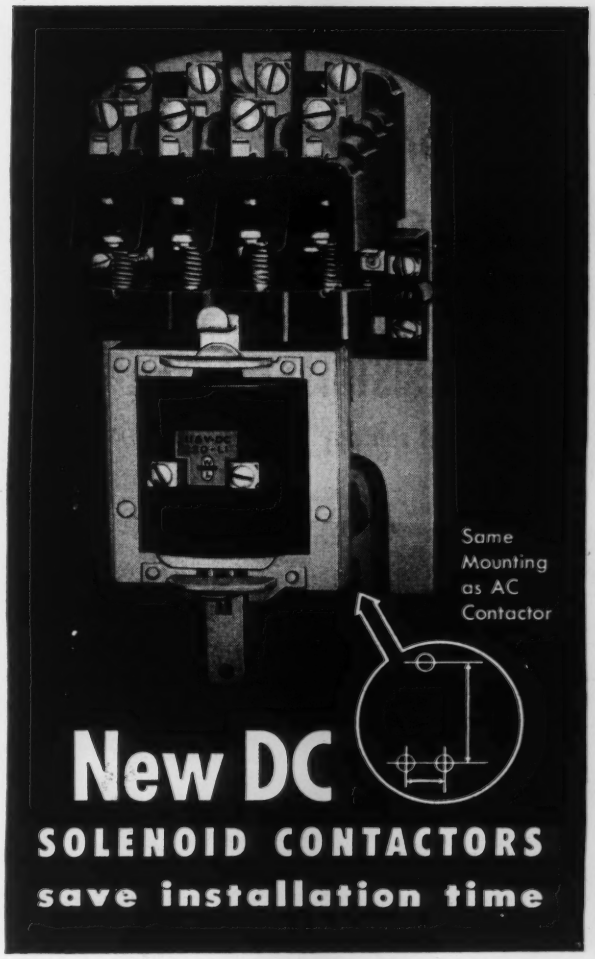
   C. W. Drake—"Synchronized Drives", *Product Engineering*, July, 1940.
3. "Synchronized Drives without Mechanical Connection", *Siemens Zeitschrift*, September, 1927, Page 603.
   Dreyfus—"The Synchronized Running of Polyphase Asynchronous Motors", *Elektrotechnik & Maschinebau*, 1930, Page 985.
4. E. L. Schwarz-Kast—Austrian Patent No. 134689, "Safety Protection Against Unpermissible Misalignments of Crane Bridges".



Unit construction—
Fit standard AC mounting

Now, with this modern, compact solenoid-type contactor . . . you'll no longer have the nuisance of drilling special holes for each component and furnishing special insulation.

Ward Leonard's new Size 1, 2 and 3 DC Solenoid Contactors are *unit-insulated*—and have metal base plates which fit AC contactor dimensions.

Same accessibility of wiring as AC contactors . . . silver-to-silver contacts . . . components interchangeable with AC contactors.

Write for new Bulletin 1950. WARD LEONARD ELECTRIC CO., 58 South Street, Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

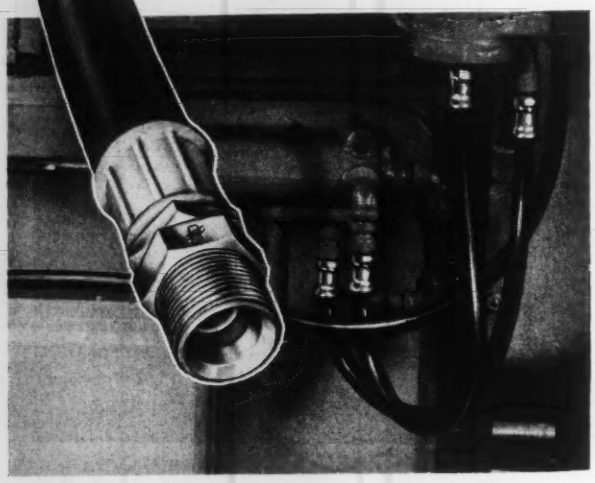
WARD LEONARD ELECTRIC COMPANY
Result-Engineered Controls
RESISTORS • RHEOSTATS • RELAYS • CONTROL DEVICES

Strong on Flexibility ...Erosion-Proof, too!

...Two strong points of

RESISTOFLEX ASSEMBLIES

that can pay off in your oil circuits



Resistoflex hose has what it takes for long service in hydraulic applications. Its capacity to endure constant flexing, vibration, shock loads—plus its remarkable inertness to oils—have proved to be the right answer for reduced troubles and replacements. That's why, for years, *Resistoflex* hose assemblies have been standard equipment in products of many well-known manufacturers.

With a high tensile compar tube, Resistoflex assemblies possess more than the necessary strength for medium as well as low pressure installations. Since this compar tube is so completely unaffected by oils, there's never any swelling, gumming or erosion—your guarantee of unclogged hydraulic cylinders and pump!

If *you* need a flexible hose assembly that can take it—one with a record of trouble-free transmission of oils, fuels, solvents, refrigerants and gases—you'll profit by equipping with Resistoflex assemblies. Write us for specific recommendations or catalog data.

Trade-Mark Reg. in U. S. & Can. Pat. Off.

RESISTOFLEX

RESISTOFLEX CORPORATION

Belleville 9, New Jersey

SYNTHETIC FLEXIBLE PRODUCTS AND PARTS FOR INDUSTRY

Hydraulic Differential Drive

(*Concluded from Page* 132)

parts. The planet carrier itself, for instance, represents a difficult machining job because of the large number of critical tolerances in a single piece. It is perhaps typical of aircraft design in this regard where reduction of the number of parts to insure operating simplicity takes precedence over simplification of individual parts for machining purposes.

APPLICATIONS OF THE DRIVE: A useful characteristic of a differential drive is the relationship between output torque and hydraulic pressure. Equation 4 shows that the effect of friction is the only disturbing factor of a linear proportional relationship between output torque and hydraulic pressure. This property could be used to build a "constant-torque" control if so desired.

Throughout the article the high-speed shaft has been referred to as the "output" shaft. This represents the relationship for the supercharger drive. However, there is no physical reason for not reversing the power flow. Here the variable-displacement motorpumps would be on the "output" shaft and the hydraulic pressure would be proportional to "input" torque. By utilizing this property in a constant-torque control, a property often sought in automotive transmissions is possible.

Many Design Variants Are Possible

For instance, if the gear ratios were varied to give a ratio range between 1 to 1 and 3 to 1 reduction, and the variable-displacement motorpumps were on the output shaft, a practicable automotive transmission would be achieved which would have the desirable property of holding the engine to a preselected torque value set by the accelerator pedal. It is well known that a constant-engine-torque drive tends to give good fuel economy.

This illustration serves to show one of a great many possible design variants. The equations are valid for any differential drive, regardless of type of gearing or ratio, if it is remembered that the subscripts 1 refer to the reactor member, subscripts 2 refer to the transmission shaft (input *or* output) directly connected to the by-pass power circuit, and subscripts *O* refer to the transmission shaft with no connection to the by-pass circuit.

Differential drives being useful and versatile devices, it is unfortunate that they are misapplied frequently. Where the ratio $R_{n\max}/R_{n\min}$ is less than 1.8 to 2.0, less complicated drives will do the job in most cases. On the other hand, when $R_{n\max}/R_{n\min}$ exceeds 3.5 to 4.0, the amount of power handled by the by-pass circuit is so large that little is to be gained by the differential principle. Inventors in this field frequently neglect the amount of by-pass power and are surprised at the final size and weight of the reactor motor system.

stainless steel is a family affair

There's more than just one type of stainless steel . . . because *stainless* applies to a family of steels. And in order to get the best possible results from stainless in your application, the *right* analysis must be used.

That's why Crucible, a pioneer in the development of this specialty, offers you the services of an alert staff of metallurgists to help you with your stainless application problem.

Crucible's half century of specialty steel leadership is built on a strong foundation of service to Industry . . . with attention to detail . . . whether the order is in tons or pounds. From the ground up, Crucible designed and put into operation one of the first integrated mills built specifically to hot and cold roll stainless steel. This $18,000,000 addition gives Crucible facilities to provide industry with stainless in every form.

There is no substitute for Crucible's background of 50 years of specialty experience. Let Crucible show you how to apply stainless steels to your products. One call from you puts us to work on your application.

CRUCIBLE STEEL COMPANY OF AMERICA, Chrysler Building, New York 17, N. Y.

CRUCIBLE first name in special purpose steels

STAINLESS STEELS

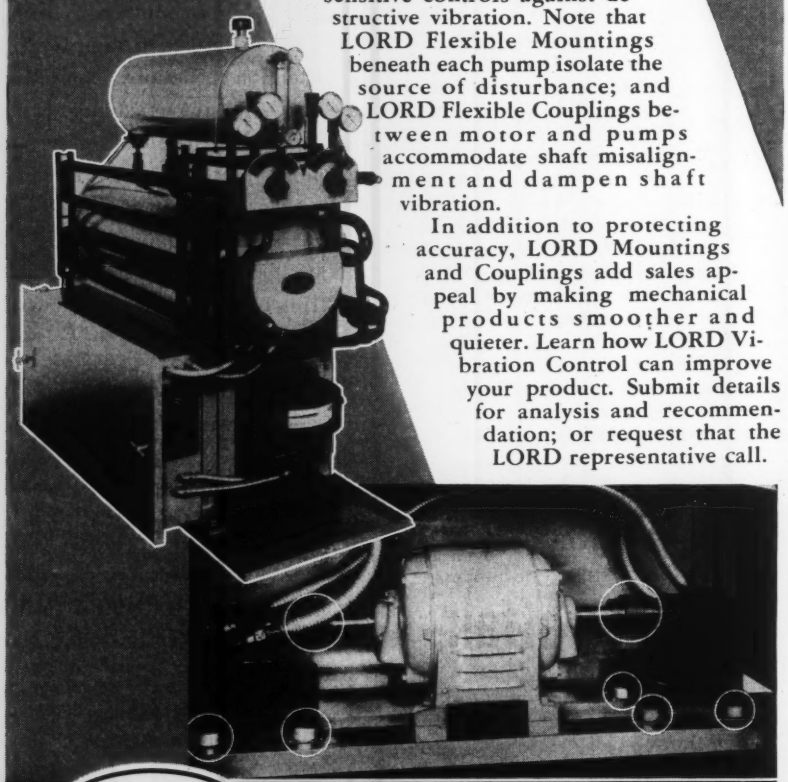
STAINLESS • HIGH SPEED • TOOL • ALLOY • MACHINERY • SPECIAL PURPOSE • STEELS

Protected with LORD BONDED RUBBER PRECISION! Mountings and Flexible Couplings

Precision of temperature control is the basis for uniform quality in many products. Typical of the equipment which automatically maintains process temperatures within close limits under varying load conditions is the Royle Temperature Control Unit shown here!

It is also typical of modern equipment design that LORD products are used to protect sensitive controls against destructive vibration. Note that LORD Flexible Mountings beneath each pump isolate the source of disturbance; and LORD Flexible Couplings between motor and pumps accommodate shaft misalignment and dampen shaft vibration.

In addition to protecting accuracy, LORD Mountings and Couplings add sales appeal by making mechanical products smoother and quieter. Learn how LORD Vibration Control can improve your product. Submit details for analysis and recommendation; or request that the LORD representative call.



LORD BONDED RUBBER Vibration Control

LORD MANUFACTURING COMPANY · ERIE, PA.

Canadian Representative: Railway & Power Engineering Corp. Ltd.

SALES AND SERVICE PERSONNEL

HAVING SERVED as manager of technical sales, with headquarters in Coatesville, Pa., for the last four years, **William G. Theisinger** has been appointed regional manager of sales for Lukens Steel Co. In his new positon, with headquarters in Houston, Tex., Dr. Theisinger will supervise the activities of Lukens district sales offices and sales representatives in twelve southern states, from South Carolina to California.

♦

An employee of the Babcock & Wilcox Tube Co. for thirty-three years, **H. F. Lefferty** has been appointed Pacific Coast sales manager for the company. His new headquarters will be in the Petroleum Bldg., 714 Olympic Blvd., Los Angeles, Calif.

♦

Graver Tank & Mfg. Co. Inc., East Chicago, Ind., has announced the appointment of **Harry A. Dennjs** as manager of sales for the company's stainless and alloy steels division. He will also continue his present assignment as manager of weldment sales. Mr. Dennis formerly was assistant district manager of sales for Lukens Steel Co. and their by-products and Lukenweld divisions.

♦

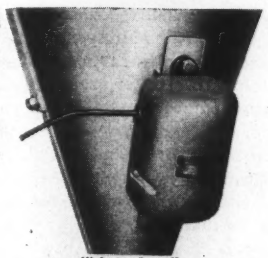
With seventeen years' sales and engineering experience in the accessory field, **R. S. Atkinson** has been appointed sales manager of the Romec division of Lear Inc. Located at Elyria, O., Mr. Atkinson will direct the sales of Lear-Romec industrial and aircraft pumps and valves.

♦

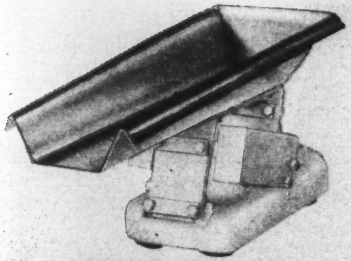
R. M. Junker has been appointed manager of the industrial roll sales department, Goodyear Tire and Rubber Co., according to an announcement from the mechanical goods division. He replaces **Ernest Peterson,** who has retired.

♦

New appointments in General Electric Co.'s transformer and allied product divisions have been announced recently. **Max I. Alimansky,** until recently engineer of the capacitor divisions, has been appointed assistant to the manager of engineering and is succeeded by **Merritt E. Scoville,** formerly assistant engineer. **Clement E. Sutton Jr.** has been



"Noiseless"
BIN VIBRATORS
Make Stubborn Materials Flow Freely



VIBRATORY FEEDERS
Finger-Tip Control of the Rate
of Bulk Material Flow

VIBRATORY PACKERS
Pack and Settle Materials
in Containers and Molds

USE COMPONENTS OF *SYNTRON* DEPENDABLE QUALITY

SMALL PARTS FEEDERS
Single Line Feed — Piece by Piece

ELECTRIC VIBRATORS, promoting free-flowing materials out of bins, hoppers and chutes.

VIBRATORY FEEDERS, controlling the flow of materials in packaging machines, batching systems, etc.

VIBRATORY PACKERS, settling materials in containers and in molds.

SMALL PARTS FEEDERS, feeding small objects to automatic machinery, inspection belts, counting and packaging machinery.

VIBRATING GRIZZLIES, for rough scalping and separation.

VIBRATING SCREENS, removing foreign objects, lumps, etc., from foods, chemicals, ceramics.

SHAFT SEALS to eliminate leakage of gases and liquids from around rotating shafts of compressors, pumps, mixers, etc.

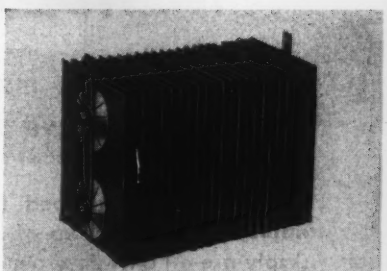
SCREW PUMPS, handling various grades of oil—crude, fuel, lube and hydraulic.

HOPPER LEVER SWITCHES, controlling the amount of material in bins and hoppers—controlling batch weighing by volume.

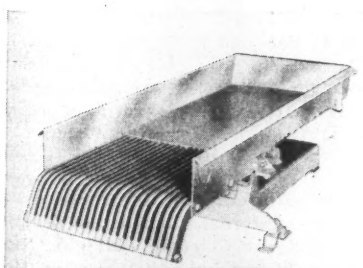
SELENIUM RECTIFIERS for efficient conversion of A.C. to D.C. in all types of electrical equipment. From 1″ sq. plates up to 6″ x 12″.

Write for catalog data

SYNTRON CO.
260 Lexington, Homer City, Pa.

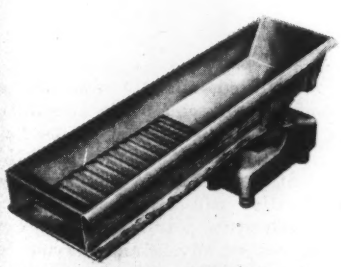


SELENIUM RECTIFIERS
As Individual Plates
or Stacks

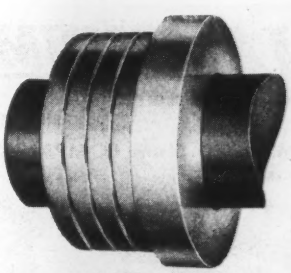


VIBRATING GRIZZLIES
Both Feed and Scalp
With One Unit

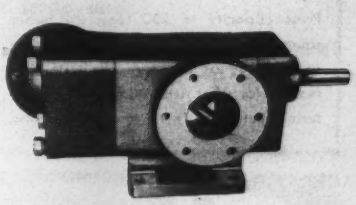
HOPPER LEVEL SWITCHES
Control Maximum and
Minimum Hopper Contents



VIBRATING SCREENS
Wet or Dry — Single or
Multiple Decks



SHAFT SEALS
Replace Packing and Eliminate Leakage
From Around Rotating Shafts



SCREW PUMPS
For Smooth, Pulseless
Continuous Axial Flow

Did you know that SHIMS can be MADE ADJUSTABLE?

[redacted]

Laminated shims, stamped from sheets of ***LAMINUM****, are **adjustable.** Laminations of .002 or .003 inch brass or steel simply p-e-e-l off with a penknife, right at the job!

HERE ARE THE ADVANTAGES:

1. No more filing or precision machining or grinding. Laminated shims can take care of increased component tolerances.
2. No fumbling or counting loose shims. The pack is *bonded* together although it peels easily.
3. No dirt, oil, grease between shim layers. Shim is always uniform in gauge, clean and smooth as glass.
4. Actually *less* compressible than stacked or one-piece shims.
5. No standby lathe or operator—no new skill required.
6. Can be fitted with babbitted lugs to prevent oil or pressure loss.

FOR PRECISION ADJUSTMENT OF SPLIT BEARINGS, GEAR MESH, END-PLAY—ACCURATE SPACING OF MACHINERY COMPONENTS.

Send today for our new data file with specifications, design factors and applications. Sample of ***LAMINUM*** included.

SEE OUR CATALOG IN SWEET'S FILE

STAMPING • GRINDING METALWORKING SERVICES

Press capacity to 100 tons, 24 inches square, shallow draw. Special equipment and variety of dies can eliminate die-making for short runs. Wide stock of materials. Let us quote on your *difficult* jobs.

**LAMINUM* (Reg. U. S. Pat. Off.) shims are solidly bonded units made up of .002 or .003 inch brass or steel laminations with a microscopic layer of metallic binder. Cut to your exact specifications.

LAMINUM

THE SOLID SHIM THAT peels FOR ADJUSTMENT

LAMINATED SHIM COMPANY, Inc.

1204 Union Street Glenbrook, Conn.

SHIMS SHIM STOCK STAMPINGS AN-COR-LOX NUTS

named manager of the distribution transformer sales division. His former position as manager of the power transformer sales division is now held by **John W. Butler,** former manager of the commercial engineering division. A new general service division has been formed with **Howard R. Humphrey,** formerly administrative assistant to the manager of sales, as manager.

♦

New sales manager for the standard contol division of Westinghouse Electric Corp. at Beaver, Pa., is **F. H. Clark,** who was formerly manager of the small a-c control sales section.

♦

William F. MacDonald has been elected president of E. F. Houghton & Co., Philadelphia manufacturer of industrial oils, leathers and chemicals. Temporarily he will continue to hold also his present office as treasurer. Mr. MacDonald succeeds **Major Aaron E. Carpenter,** who has been elected chairman of the board of directors and who will remain the chief executive officer of the company.

♦

Having been active in the development of welding for the past twenty years, **C. L. Stocker** has assumed sales and engineering responsibilities for Lincoln Electric Co. in the San Francisco area. The welding engineering staff of the San Francisco area office, located at 1303 Stanford Ave., Emeryville, Calif., has recently been increased by the addition of **M. F. Yale** and **T. Nichol.** Also, **A. L. Patnik** has been given responsibility for Lincoln welding engineering in the Seattle district, with offices at 1914 Utah Ave., Seattle, Wash.

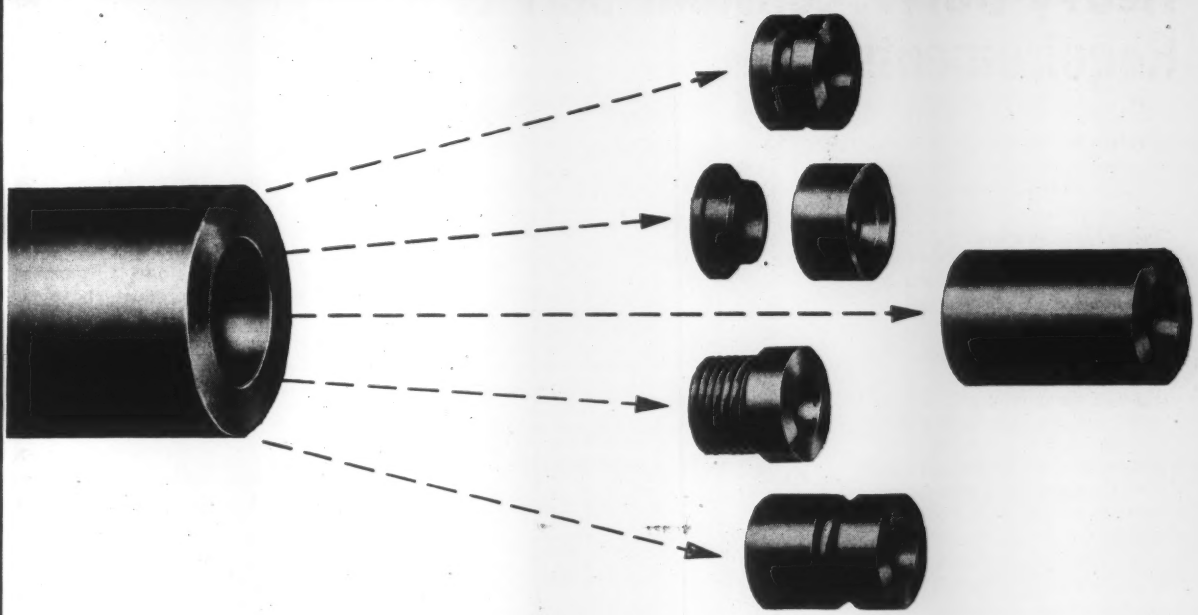
♦

At a meeting of the membership of the Internal Combustion Engine Institute at Chicago, **Philip A. Norton,** sales manager of Wisconsin Motor Corp., Milwaukee, was elected president for the ensuing year. He succeeds **H. W. Smith,** manager of engine sales, Caterpillar Tractor Co., Peoria, Ill.

♦

Announced recently was the appointment of **John North** as manager of the St. Louis office of the De Laval Steam Turbine Co. of Trenton, N. J. Operating as a direct factory branch, this office will be in a position to give industrial plants in the St. Louis area expert advice and service concerning centrifugal pumps, centrifugal blowers, steam turbines, speed reducers and other equipment manu-

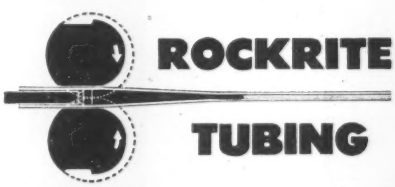
ROCKRITE TUBING SAVES UP TO 50% OF MACHINING COSTS ON THESE PARTS



You may be saving money by machining parts like those shown here from ordinary mechanical tubing instead of from bar stock or forgings. But with lighter close-tolerance Rockrite Tubing, cost-saving possibilities are still greater! Here's why:

1 Rockrite Tubing is sized by a distinctively different process to much closer tolerances than are practicable by any other method.

2 Machine output has been doubled and machining costs cut in half when Rockrite was substituted for standard mechanical tubing.



LEADS ALL OTHERS IN THESE SAVINGS

- Higher cutting speeds
- Tools last longer between grinds
- Work-surface finishes are better
- Stations on automatics are often released for additional operations
- Extra-long pieces available — less downtime for magazine stocking and fewer scrap ends
- Closer tolerances often eliminate necessity for machining on outside or inside

WANT TO KNOW THE 3 REQUIREMENTS ESSENTIAL FOR TUBE ACCURACY AND COST SAVINGS? Send for new Bulletin being prepared on "behind-the-scenes" facts on close-tolerance Rockrite Tubing. TR-114

TUBE REDUCING CORPORATION · WALLINGTON, NEW JERSEY

AT LAST!
A Two-Way D. C. Poppet-Type Solenoid Valve *That's Engineered* to Meet Heavy-Duty..Constant Service Requirements

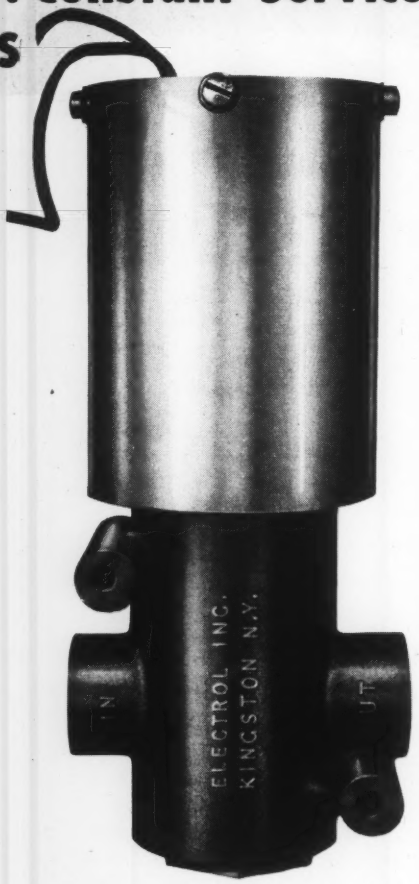
SIMPLE in design — and engineered to meet heavy-duty . . . constant service requirements —the new Electrol Two-Way D. C. Poppet-Type Solenoid Valve is winning wide acceptance among design engineers, who are aiming to improve a product and lower its operating cost.

It is available in either *normally open* or *normally closed* types, and can be used in oil, air or water, with pressures up to 1500 P.S.I. The Solenoid is designed to withstand constant energizing, and is obtainable in voltage increments ranging from 6 to 220-volts D. C.

The Valve is of the Poppet Type to assure an *absolute minimum of internal leakage*. It uses standard hydraulic packings, and has two ½" NPT ports, with a maximum rated flow of 5½ gallons per minute. Reducing bushings are supplied where NPT ports of ⅜", ¼", or ⅛" are required.

The Valve measures 8-3/32" x 3½" x 3", weighs approximately 10 lbs., and has two mounting bosses with holes which will accept ¼" bolts.

When ordering—specify maximum operating pressure, type of fluid used, voltage, port size, the flow required in gallons per minute, and whether the Valve is to be *normally open* or *normally closed*.



Our engineers will be glad to work with you in adapting this unit to meet your specific requirement—or on any problem involving the use of hydraulic devices.

Better Designed Products Use Electrol Hydraulics

Electrol
INCORPORATED
KINGSTON, NEW YORK
CYLINDERS · SELECTOR VALVES · FOLLOW-UP VALVES
CHECK VALVES · RELIEF VALVES · HAND PUMPS
POWERPAKS · LANDING GEAR OLEOS · SOLENOID
VALVES · ON-OFF VALVES · SERVO CYLINDERS · TRANSFER
VALVES · CUT-OUT VALVES · SPEED CONTROL VALVES
FOR BETTER HYDRAULIC DEVICES

factured by the company. At the same time it was announced that **H. H. Reynolds** has been appointed manager of the company's Los Angeles office. Having been located in the home office of the company for many years, Mr. Reynolds is intimately familiar with the engineering design and application of all De Laval products.

♦

Oliver J. R. Troup Jr. has been named sales engineer for the Atlas Chain & Mfg. Co. Well versed in the application of roller chain, Mr. Troup has been assigned the territory of eastern Pennsylvania, New York State and Maryland.

♦

Recently organized to take over the Electrimatic line of refrigeration fittings, Madden Brass Products Co. has appointed **F. H. Moss** as sales manager. Mr. Moss, formerly assistant to **M. B. Madden,** who heads up the new line, has a thorough knowledge of the products he represents. Both Mr. Madden and Mr. Moss are well known to the trade, having been key men with Electrimatic division of the Simonize Co.

♦

Aeroquip Corp. of Jackson, Mich. has announced that **Byron E. Snow** has been appointed sales engineer in the Chicago territory, including Illinois, Iowa and southeastern Wisconsin. He will service only the manufacturing accounts in that territory, and his headquarters will be 380 Elizabeth Drive, Lombard, Ill.

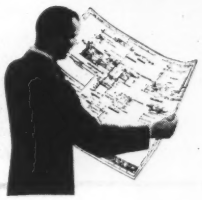
♦

William B. McFerrin has been appointed division executive vice president, and **Robert M. Briney has** been made division vice president in charge of wrought alloy products, Haynes Stellite division of Union Carbide and Carbon Corp., Kokomo, Ind. Mr. McFerrin had been assistant manager of development, electro metallurgical division, Detroit, and Mr. Briney was manager of the development division, Union Carbide and Carbon Research Laboratories Inc., New York.

♦

Announced recently was the promotion of **Frank R. Freyler** to the position of manager of Allis-Chalmers' mid-Atlantic region. Mr. Freyler, who has been with the company for twenty-one years and most recently was Philadelphia district manager, will continue to have his headquarters in Philadelphia. He succeeds **William Arthur,** who is soon to retire, but before taking full retire-

Designed for industrial microfilming requirements



...Kodagraph Micro-File Equipment

TO CONDENSE, PRESERVE, PROTECT YOUR DRAWINGS...THE KODAGRAPH MICRO-FILE MACHINE

Place an engineering drawing on the copyboard . . . touch a foot-switch . . . and the machine does the rest—

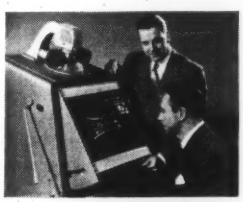
Instantaneously, it takes the picture at great reduction . . . advances the film . . . is ready for the next drawing.

In this manner you can record documents—large or small—bound or unbound. Do it conveniently . . . speedily . . . whenever the demand arises.

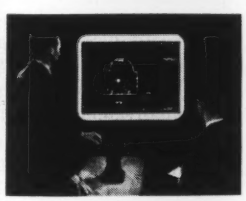
4 different models now available

Kodagraph Micro-File Machines meet the wide range of requirements in engineering departments . . . and in the field. Each is a precision machine . . . designed by Kodak . . . made by Kodak to give you these advantages—

- You can reclaim 98% of your filing space . . . microfilm old records . . . then destroy them, if you wish.
- You can provide branch offices, repair depots with compact, duplicate records on positive microfilm.
- You can simplify reference work . . . file your 16mm. or 35mm. film records at your finger tips . . . ready for immediate review in the Kodagraph Film Reader.



To view your microfilm records in the office . . . the Kodagraph Film Reader. Each image is enlarged sharp and clear . . . easy to read and check . . . on a special green-tinted viewing screen. You can speed the film from image to image . . . get all the information you need—quickly.



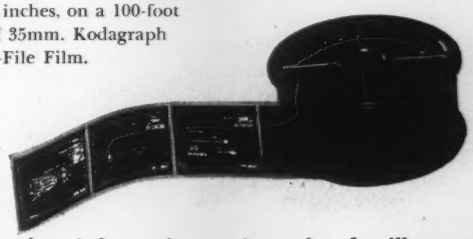
To view your microfilm records in the field . . . the Kodagraph Portable Projector. You can flash your records on any screen or light colored wall . . . magnified at dimensions up to several times larger than the original, if necessary. It is convenient to carry—weighs only twelve pounds.

To make enlargement prints from your microfilm records . . . the Kodagraph Micro-File Enlarger. It produces prints of maximum legibility . . . so sharp that they can be used as intermediates in direct process or blueprint machines.

"Kodak" is a trade-mark

EASTMAN KODAK COMPANY
Industrial Photographic Division
Rochester 4, N. Y.

In weighing the advantages of installing a Kodagraph Micro-File Machine in your engineering department . . . consider, too, the economy of the operation. You can record 675 drawings, size 24x36 inches, on a 100-foot roll of 35mm. Kodagraph Micro-File Film.



For complete information, write today for illustrated literature.

19

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Rochester 4, N. Y.

Please send your literature illustrating Kodagraph Micro-File Equipment.

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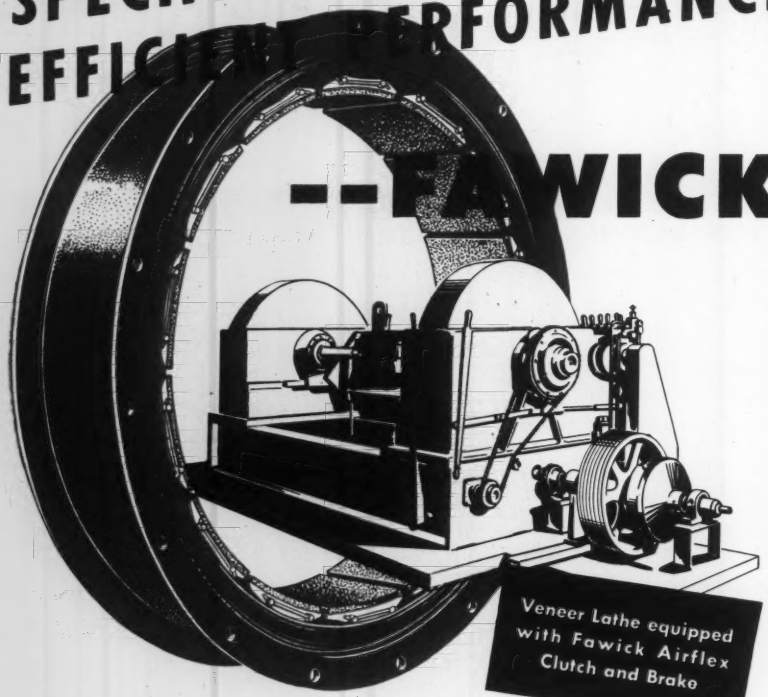
Kodak

SPECIFICATION for EFFICIENT PERFORMANCE --FAWICK



Veneer Lathe equipped with Fawick Airflex Clutch and Brake

Machines of many types and sizes are constantly proving the efficiency of Fawick Airflex Clutch and Brake units in their operation.

When the clutch operating requirements for any machine call for (1) quick response; (2) complete and sensitive control; (3) shock-free operation; (4) long, maintenance-free service—specify Fawick.

Fawick design and operating features provide clutch or brake action by 360° "constant-velocity" contact on the full width of the friction surface. Torque is transmitted through the flexible rubber tube, cushioning peak shock loads and dampening shaft vibrations.

Add to these the features of complete torque control and remote operating control and you have the component features for top-efficiency clutch or brake action—Fawick makes good machines perform better.

For specific information on all advantages of Fawick Clutch and Brake units, write to the Main Office, Cleveland, Ohio, for Bulletin 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK Airflex CO., INC.

9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = AXIAL CONTACT + 360° RADIAL CONTACT + SHOCK ABSORPTION + CONTROLLED TORQUE = PEAK EFFICIENCY

ment will serve as special representative for the mid-Atlantic region. Territory included in this region is covered by district offices in Baltimore, Charleston, Philadelphia, Richmond, Wilkes-Barre and York. **Arthur D. Brown,** who has been manager of the firm's Washington, D. C., office since 1947, succeeds Mr. Freyler as manager of the Philadelphia district office, and **R. N. Landreth,** associated with the Washington office since 1934, is the new manager of that office.

♦

Huck Manufacturing Co., Detroit, has recently announced the promotion of **Robert N. Hendrickson** to vice president in charge of sales engineering and the appointment of **Frank A. Dobbe** as vice president in charge of sales.

♦

Promotion of **W. L. Tomlinson** to the position of manager of automotive glass sales for Pittsburgh Plate Glass Co. was announced recently. Associated with the company since 1928, Mr. Tomlinson has served as Detroit district manager of industrial glass sales during the past seven years. He will maintain his headquarters at 6045 Hamilton Ave., Detroit, Mich.

♦

Having been associated with the plastics industry for several years, **John L. Goodrow** has been appointed sales manager of the custom molding division of Rogers Plastic Corp., West Warren, Mass.

♦

Telechron Inc., Ashland, Mass., has announced the appointment of **Walter F. Greenwood** as manager of industrial sales. Mr. Greenwood comes to Telechron from another General Electric affiliate, Trumbull Electric Mfg. Co., where he had been works manager of the Los Angeles and San Francisco branch plants since 1947. Prior to that he was sales manager of the dynamometer division, Clayton Mfg. Co. of California.

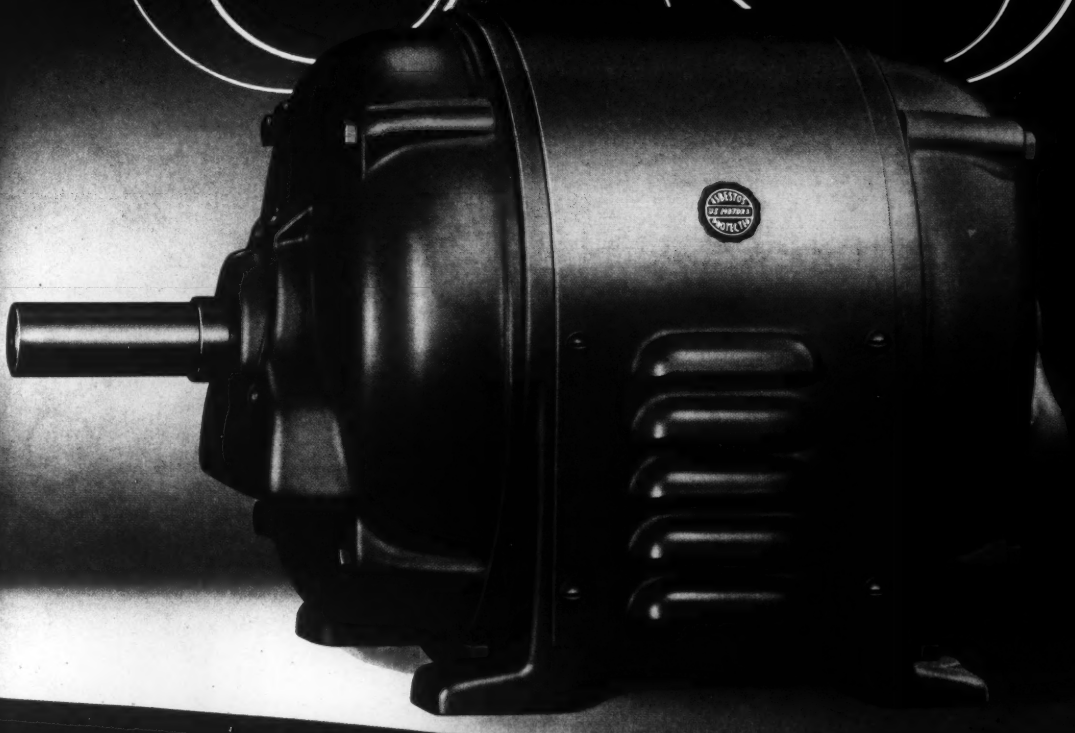
♦

James W. Elizardi has recently been made manager of the Houston, Tex. sales office of the Marley Co. Inc. of Kansas City, Kans.

♦

Having served for ten years as manager of the Chicago office and warehouse of the American Screw Co., Providence, R. I., **Arthur L. Fleck** has been named general sales manager of the company, succeeding the late **Charles O. Drayton.**

U S UNICLOSED MOTOR



. . the Armored Motor

COMPLETELY SHIELDED

U S MOTORS

(See next page)

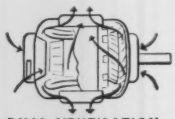
U. S. UNICLOSED MOTOR

1 **ARMOR-CLAD WINDINGS**
Asbestos Protected

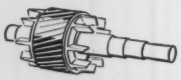
2 **LONGER BEARING LIFE**
With Lubriflush

3 **NORMALIZED CASTINGS**
Maintain Accuracy

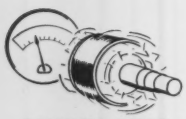
4 **UNICLOSED CONTOURS**
Give Extra Protection



5 **DUAL VENTILATION**
For Ever-Cool Operation



6 **SOLID CENTRICAST ROTOR**
With Integral Fans

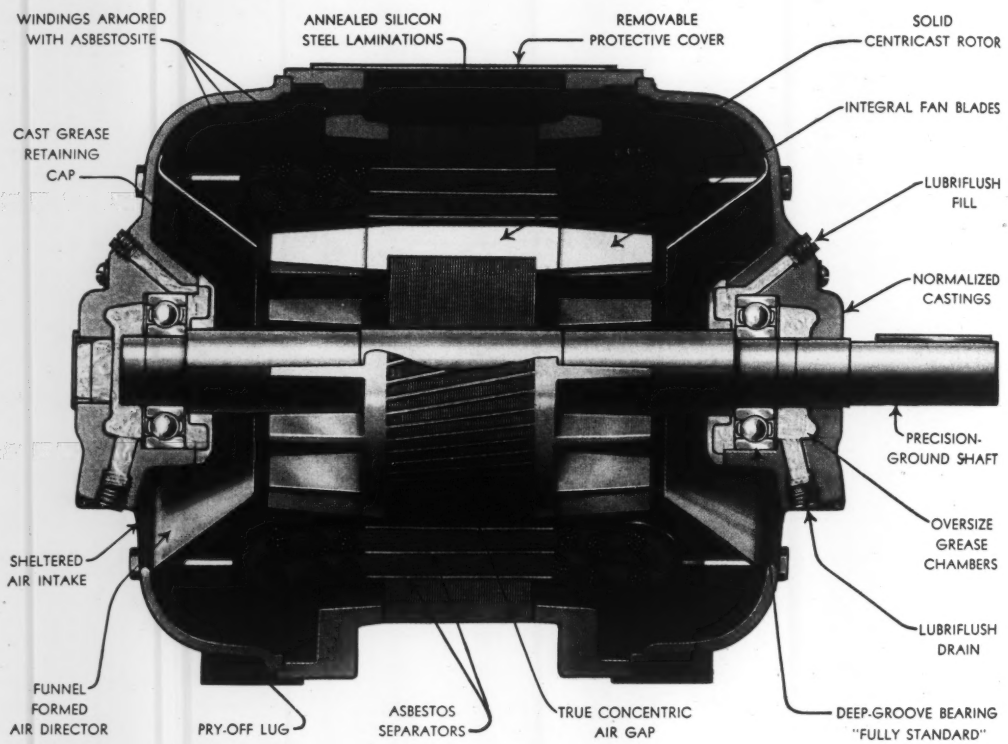


7 **BALANCED ROTOR**
Eliminates Vibration

8 **ANNEALED LAMINATIONS**
For Lasting Economy

9 **PRECISION MACHINING**
Matches Bearing Accuracy

10 **GUARANTEED SATISFACTION**
Nation-Wide Service



THE MOTOR THAT GIVES YOU MANY PLUS VALUES

The Uniclosed is no ordinary motor. It combines external protection with interior perfection. The vital operating parts are armored against damage from moisture, falling debris, dust and dirt. The Uniclosed is suitable for many conditions ordinarily requiring more expensive splash-proof or enclosed designs. Asbestos-protected windings guard against carbonization. Lubriflush lubrication insures longest life of bearings which are *self-lubricated* for normal life. Normalization of all castings insures permanent bearing alignment. Dual cyclone ventilation insures a cool running motor in high atmospheric temperatures. The rotor is centricast. The steel laminations are annealed to prevent excessive electrical loss.

Request these two interesting, informative bulletins.

Write today for Bulletins featuring U. S. Uniclosed and U. S. Sanitary motors. These products are illustrated in natural colors. Details of construction reveal many features of interest to cost-minded plant operators seeking the most dependable economical power. If your power requirements include variable speed, ask for Bulletin featuring U. S. Varidrive Motors.

U.S. ELECTRICAL MOTORS Inc.

PACIFIC PLANT: Los Angeles 54, California **ATLANTIC PLANT: Milford, Connecticut**

Atlanta 3, Ga.; Bakersfield, Calif.; Boston 16, Mass.; Chicago 8, Ill.; Cincinnati 16, Ohio; Cleveland 14, Ohio; Dallas 9, Texas; Detroit 2, Mich.; Fresno 1, Calif.; Houston 4, Texas; Indianapolis 4, Ind.; Milwaukee 2, Wisc.; Minneapolis 2, Minn.; New York City 6, N. Y.; Philadelphia 2, Pa.; Pittsburgh 2, Pa.; San Francisco 7, Calif.;
Distributors and Agents in all principal cities.

XUM

SALES NOTES

HERETOFORE the **Curtis Universal Joint Co.,** Springfield, Mass., has been represented exclusively in the United States by the **Boston Gear Works** of North Quincy, Mass. A new policy enables any user of industrial universal joints to order direct from Curtis, thereby assuring faster delivery and enabling a customer to specify special hub borings or machining. Users or designers can avail themselves of a complete engineering service by sending prints of installation, together with torque speed, load and angle of operation, direct to the Curtis plant.

♦

Twenty-six thousand square feet of additional office and warehouse space in Detroit, Mich. have been purchased by the **Minnesota Mining & Mfg. Co.** Moving the branch unit to the new quarters at 8825 Grinnell Ave. will make more manufacturing space available in the company's adhesives plant at 411 Piquette Ave.

♦

Syntron Co., Homer City, Pa., has announced the removal of its New York sales and engineering office from its old address in Long Island City to 1860 Broadway, New York 23, N. Y.

♦

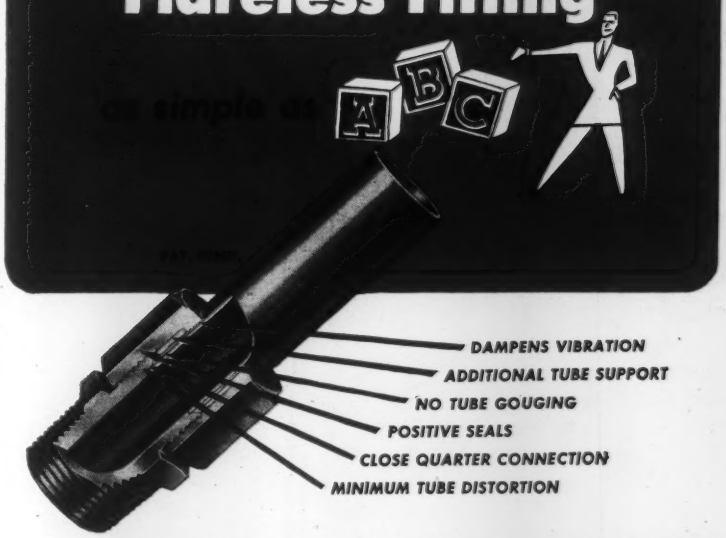
Two new sales representatives have been appointed by the **Precision Tube Co.,** Philadelphia, to handle their line of small seamless mechanical tubing. In the Chicago area, the company's new representative is **Otto Bussenius and Co.,** 80 East Jackson Blvd., Chicago 4, Ill., and in Ohio, the **C. E. Anderson Co.,** 4500 Euclid Ave., Cleveland 3, O.

♦

In a move to enlarge its service to motor customers throughout the country, the **A. O. Smith Corp.** has contracted for new manufacturing and supply facilities and has set up a more flexible sales organization. The company has contracted with **Whirl-A-Way Motors Inc.,** Tipp City, O., to manufacture Smithway fractional horsepower motors to specification. These facilities will serve the eastern region of the United States. The western region of the country will continue to be served by the

From the nation's largest exclusive manufacturer of fittings

a NEW all-purpose Flareless Fitting



IMPROVED DESIGN of the AFCO flareless fitting eliminates tube gouging and limits tube distortion — permits quick economical installation without special tools, requires no tube flaring, and is ideally adaptable to close quarter connections.

LEAK-PROOF — PRESSURE-PROOF — the gripping action of the fitting sleeve forms a positive leak-proof seal and makes the fitting especially suitable for thin wall tubing of brass, copper, steel, stainless steel, or aluminum alloy.

PRECISION MADE — AFCO fittings are available in all popular shapes for tube sizes up to and including 1 inch. Special sizes and shapes for tubing up to two inches O.D. can be furnished. All "AFCO" fitting shapes (elbows, tees, etc.) are machined from forgings — all other parts are machined from bar stock.

MEETS ALL SPECIFICATIONS — The AFCO flareless tube fitting meets all requirements of the J. I. C. Hydraulic Standards for industrial equipment and Specification AN-F-47. Because of close tube fit, the AFCO Flareless Fitting is particularly desirable in connection with food processing where sanitation and cleanliness are so vital.

SPECIALISTS IN FLARELESS • FLARED • PIPE • AND HOSE FITTINGS

AFCO *Fittings*

The Aircraft Fitting Company
1400 East 30th Street • Cleveland 14, Ohio

SEND FOR YOUR FREE COPY OF BULLETIN F-10

LARGEST PRODUCER IN THE COUNTRY
ENGAGED EXCLUSIVELY IN THE MANUFACTURE OF FITTINGS

Miller

High Pressure Hydraulic Cylinders

met J.I.C. HYDRAULIC STANDARDS years before their adoption

UNABRIDGED COPY of "Joint Industry Conference" Standards for specifying "quality" hydraulic equipment SENT FREE ON REQUEST

Why Experiment? . . . Specify Miller "Time Tested and Proven" Cylinders—And Be SURE!

All Miller Standard High Pressure Hydraulic Cylinders met both the *mandatory* and *recommended* practices of the "Standards" *years ago.*

Some "Standards" specifications, such as dirt protectors, scratch-resistant piston rods, etc., are required only under severe conditions. *All* Miller *Standard* High Pressure Hydraulic Cylinders met *these* requirements *years ago.*

The desire for elimination of *manual* rod seal adjustment is strongly voiced at all hydraulic industry conferences. The *"time tested and proven"* Miller Patented Rod Seal is self-adjusting and wear compensating . . . *requires no manual adjustment.*

Complete Line

- AIR CYLINDERS 1½" to 20" Bores
- LOW PRESSURE HYDRAULIC CYLINDERS 1½" to 14" Bores
- HIGH PRESSURE HYDRAULIC CYLINDERS 1½" to 12" Bores

Single and Double Acting, Single and Double Rod End, Spring Return, Cushioned and Non-Cushioned, Over-Sized Rod, and Long Stroke Cylinders in All Mounting Styles.

write for

Miller Air Cylinder Bulletin A-105 and Miller Hydraulic Cylinder Bulletin H-104

MILLER MOTOR COMPANY

4025 N. KEDZIE AVE. • CHICAGO 18, ILLINOIS

AIR AND HYDRAULIC CYLINDERS · ACCUMULATORS · COUNTERBALANCE CYLINDERS · BOOSTERS · AIR HOISTS

Miller

company's present Los Angeles motor manufacturing plant. Also, A. O. Smith has appointed D. L. Mills, president of Whirl-A-Way, as Eastern regional sales manager of the motor division, with headquarters in Dayton, O. Three Eastern area sales managers report to Mr. Mills. They are Hal E. Rowland with headquarters in Dayton, J. P. Hoffberger in New York, and R. O. Dehlendorf in Chicago. Mr. Hoffberger continues as the company's New York area sales manager.

♦

A new address has been announced by the **Electro Switch Corp.** Renaming and redefining of some of the suburban streets has changed the company address to 167 King Ave., Weymouth 88, Mass.

♦

C. P. Clare & Co., Chicago relay manufacturers, has announced the appointment of two new sales representatives. **Maury E. Bettis Co.,** 3119 Gillham Rd., Kansas City, Mo., will cover the territory of Kansas, northern Oklahoma and western Missouri, and **Norman W. Katherinus & Co.,** 1218 Olive St., St. Louis, Mo. will represent the company in eastern Missouri.

♦

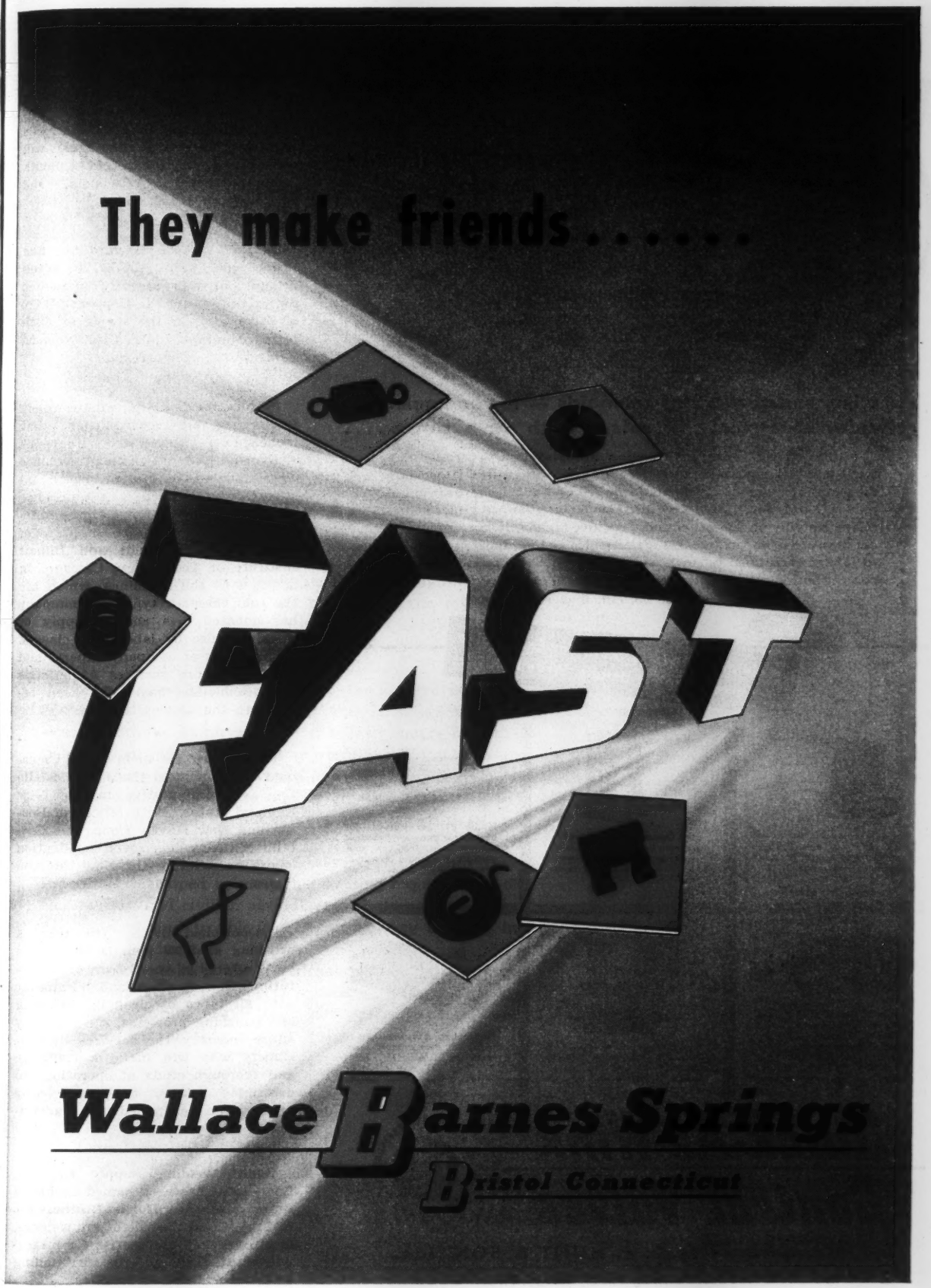
Through the acquisition of the **Gotham Instrument Co. Inc.,** New York, **American Machine and Metals Inc.** has established a new division. The new Gotham Instruments division will supplement the line of pressure, temperature, electrical, flow and liquid-level indicating instruments made by the United States Gauge division, by the addition of a complete line of industrial recording and controlling instruments.

♦

An expansion program involving a move to larger quarters has been announced by the **Follansbee Steel Corp.** for its warehouse in Rochester, N. Y. The new warehouse will carry a more extensive line of products to supplement present stocks and will be located at 961 Lyell Ave.

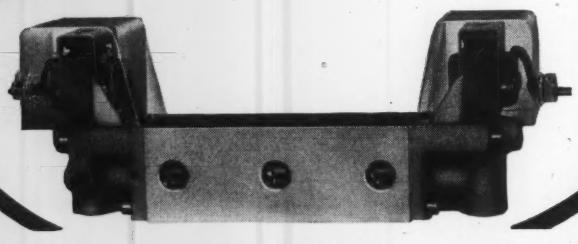
♦

Several new agents and distributors have been appointed by the **Flexitallic Gasket Co.,** Camden, N. J. The **Moorlane Co.,** Tulsa, Okla., with branches in Kansas City, Amarillo, Denver, Oklahoma City and Wichita, has been named as a new distributor. **Boiler Equipment Service Co.,** 686 Greenwood Ave., N. E., Atlanta, Ga., is a new agent serving Alabama, Florida, Georgia, North Carolina, South Carolina and Tennessee, **Lane**


They make friends.....
FAST
Wallace Barnes Springs
Bristol Connecticut

another NEW Quick-As-Wink development

Type "DS" double solenoid pilot operated single plunger control valve



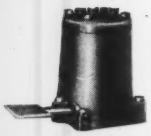
▶ smaller, shorter stroke solenoids — lower amperage
▶ simpler electrical controls — high air economy...

● Small solenoids traveling through only a 1/8" stroke move small pilot valve plungers to apply air to operating pistons that move the main valve plunger back and forth. All operating parts are rugged, yet weigh only a few ounces. Reduced amperage eliminates intermediate relays, simplifying control circuits. Positive high cycle operation, without destructive impact, reduces maintenance and minimizes trouble due to voltage variations. 3/8" to 2" sizes. 2, 3, and 4-way actions. Solenoid control both directions or automatic return in one direction.

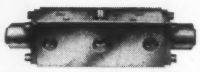
Send for Data Sheet No. 1611. It gives full details.

Individual DATA SHEETS for Each Valve — give complete details. Write today!

Hand Operated Air Valves—wide variety of uses. 2-way, 3-way, 4-way neutral position and compound exhaust.

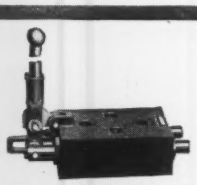


Foot Operated Air Valves—workman has both hands free, speeding production. 2-way, 3-way and 4-way actions.

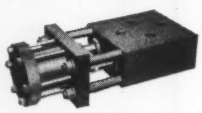


Single Plunger Valves—for air or low pressure hydraulic service. Lever, pilot, cam, diaphragm or solenoid operated. 2-way, 3-way, 4-way actions.

Series "O" and "OE" Valves—for air or hydraulic service up to 125 PSI. Push-pull, cam, pilot, diaphragm and solenoid operated. 1/8" and 1/4" pipe connections. 2-way, 3-way, 4-way and 5-way actions.



Hydraulic Valves—Up to 5000 PSI. Conservatively rated. 1/2", 3/4", 1" and 1 1/2" sizes. 2-way, 3-way, 4-way actions.



Hydraulic Valves—Up to 5000 PSI. Pilot cylinder operated. 1/2", 3/4", 1", 1 1/2", 2", 2 1/2", 3" and 4" sizes. 2-way, 3-way, 4-way actions.

Quick-As-Wink Control Valves

THE HIGHER THE PRESSURE THE TIGHTER THE SEAL

Manufactured by C. B. HUNT & SON, Inc.

1942 East Pershing Street, Salem, Ohio

Machinery Co. Inc., Buder Bldg., Seventh and Market Sts., St. Louis 1, Mo., as an agent, will serve eastern Missouri, southern Illinois, western Kentucky, western Tennessee, and northeastern Arkansas. **Plant Equipment.**, 607 Sexton Bldg., Minneapolis 15, Minn., newly appointed agent, will serve Minnesota, North Dakota, South Dakota, western Wisconsin, and northwestern Michigan.

♦

Located at 1060 Howard St., San Francisco 3, Calif., the **W. P. Wooldridge Co.** was recently appointed representative for the **Heppenstall Co.** of Pittsburgh in the states of California, Montana, Idaho, Utah, Nevada, Arizona and New Mexico.

♦

Doehler-Jarvis Corp., producer and finisher of die castings, last year launched a "university program." Fully equipped die casting machines were donated to Massachusetts Institute of Technology, Cambridge, Mass.; Rensselaer Polytechnic Institute, Troy, N. Y.; Case Institute of Technology, Cleveland; and Illinois Institute of Technology, Chicago. In addition to the machines, which are the cold chamber type, the company has donated dies and a supply of metal for use by laboratory classes. The services of company experts and periodic lecture services of officials and technicians have also been offered to the universities.

♦

The **Electric Products Co.** of Cleveland has appointed the **J. J. Costello Co.** as representative in the New England states. With offices in Boston and New Haven, Conn., the Costello company has seven application engineers covering the New England states.

♦

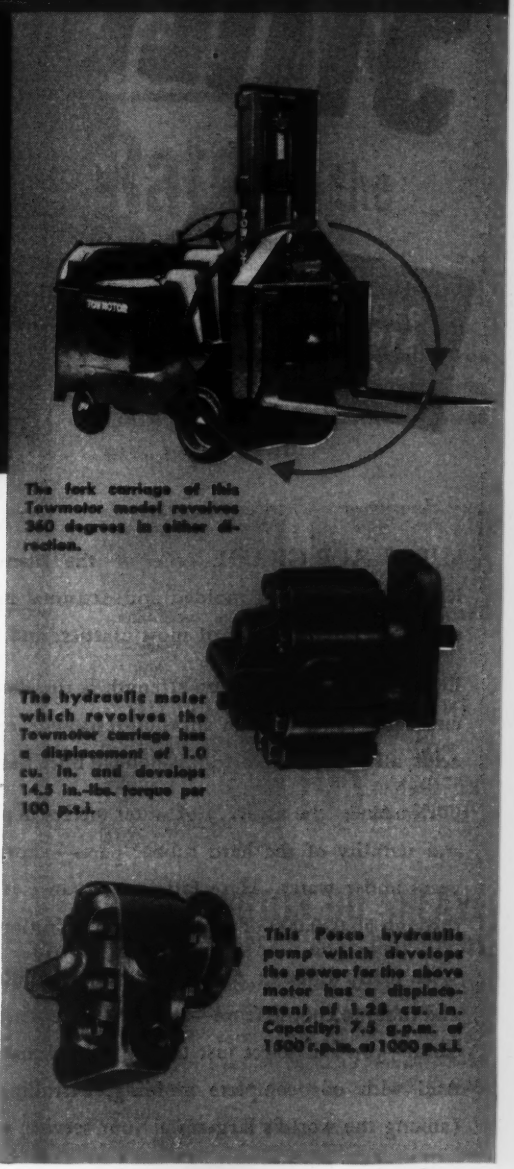
Discontinued during the war, a brushing analysis plan is once more being offered by the **Osborn Manufacturing Co.** of Cleveland. Familiar with all types of cleaning, finishing and polishing problems, the company offers industry the services of specialists who will make a complete and thorough study of operations to determine where, how, and why brushes can be applied to an advantage.

♦

White Welding Supply Co., St. Louis, Mo., has been named exclusive representative for **Hobart Brothers Co.** of Troy, O., in the sale of arc welders, electrodes, and accessories for metropolitan St. Louis and the surrounding Missouri and Illinois counties.

Another Job for Pesco HYDRAULIC MUSCLES

DUMPING MULTI-TON LOADS IN 5 SECONDS



The fork carriage of this Towmotor model revolves 360 degrees in either direction.

The hydraulic motor which revolves the Towmotor carriage has a displacement of 1.0 cu. in. and develops 14.5 in.-lbs. torque per 100 p.s.i.

This Pesco hydraulic pump which develops the power for the above motor has a displacement of 1.25 cu. in. Capacity: 7.5 g.p.m. at 1500 r.p.m. at 1000 p.s.i.

● One of industry's most effective material handling time-savers is Towmotor's fork lift truck with the revolving carriage. With it a man can quickly and easily transfer heavy loads of solid, liquid, or granular material from one container to another.

The ingenious device that revolves the fork carriage in a 360-degree circle, in either direction, is actuated by a Pesco hydraulic motor so small it will fit snugly into the palm of your hand. Yet it is so powerful it will flip loads up to three tons completely over in five seconds!

Towmotor is one of the rapidly growing number of industrial equipment manufacturers who are turning to Pesco for answers to their hydraulic problems. They have found that the skills and precision craftsmanship that Pesco has developed to meet the exacting requirements of the military and civilian aircraft manufacturers insure the high quality and trouble-free operation they demand in their own products.

Perhaps the sales power of hydraulic power can help you, too. If you have a problem in squeezing, gripping, lifting, holding, or turning heavy loads with precise accuracy and finger-tip control, why not let Pesco engineers suggest a solution.

A preliminary discussion will involve no obligation.

engineering Pesco production

PRODUCTS DIVISION

BORG-WARNER CORPORATION

24700 NORTH MILES ROAD BEDFORD, OHIO

How's this for STRENGTH?

SUPER ACE GRADE—one of the many formulas of Ace Hard Rubber available for molded and extruded machine parts, has a tensile strength exceeding that of most plastics, and high impact strength, too!

Even more important than strength is the *durability* of hard rubber—its unusual toughness, high abrasion-resistance, excellent resistance to water, acids, alkalies, etc. Only glass-bonded mica has lower moisture absorption.

For instance: the accuracy of water meters depends on strength, long-life, and stability of the hard rubber parts—expected to last 10—even 20—years under water. Here fatigue resistance is vital, as the parts may go through 50,000,000 or more cycles in a lifetime.

With many different Ace Hard Rubber molding compounds, sheets, rods and tubes to choose from—also other Ace plastics such as Ace-Tex, Parian, Saran—you can select just the right combination of technical properties. And with our complete molding, extruding and fabricating facilities (among the world's largest) at your service, your job will be done the best way, the quickest.

Always check your Ace Handbook, pgs. 4 and 5, when selecting molding materials. If you haven't a copy of this valuable 60-pg. manual, *write today—it's free.*

ACE
Since 1851

HARD RUBBER and PLASTICS

AMERICAN HARD RUBBER COMPANY

11 MERCER STREET • NEW YORK 13, N. Y.

MEETINGS AND EXPOSITIONS

Apr. 17-19—

Society of Automotive Engineers. Aeronautic meeting and aircraft engineering display to be held at the Statler Hotel, New York, N. Y. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Apr. 20-21—

Society for Advancement of Management. Annual Time Study and Methods Conference jointly sponsored by the SAM and ASME will be held at the Statler Hotel, New York, N. Y. Additional information may be obtained from Conference headquarters, Society for Advancement of Management, 84 Williams St., New York, N. Y.

Apr. 24-26—

American Society of Mechanical Engineers. Process Industries Conference to be held at the William Penn Hotel, Pittsburgh, Pa. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Apr. 24-27—

American Management Association. 19th national packaging exposition to be held at the Navy Pier, Chicago, Ill. Edward K. Moss, 330 West 42nd St., New York 18, N. Y., is public relations director.

Apr. 25-26—

Metal Powder Association. Sixth annual meeting to be held at the Book-Cadillac Hotel, Detroit, Mich. Additional information may be obtained from headquarters of the society at 420 Lexington Ave., New York 17, N. Y.

May 8-12—

American Textile Machinery Exhibition to be held in Atlantic City Auditorium, Atlantic City, N. J., under the sponsorship of the National Association of Textile Machinery Manufacturers. E. Kent Swift, Whitinsville, Mass., is president.

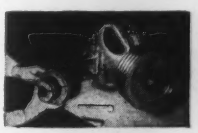
May 8-19—

British Industries Fair to be held at Olympia and Earls Court, London; and at Castle Bromwich, Birmingham, England. Additional information may be obtained from British Information Services, 30 Rockefeller Plaza, New York 20, N. Y.

How Does Your Drive Rate on the V-drive Scoreboard?

Here are 15 money-saving features of the most efficient V-belt drive. Only Worthington Multi-V-Drives give you all 15!

These Features Save You Money	For These Reasons	It's a Worthington—
1 Two-piece QD sheave—rim and hub separate	Mount light hub first, then slide heavier rim on tapered hub	EXCLUSIVE
2 Choice of one or two-piece assembly	Mount hub and rim separately, or loosely assemble hub on rim and slide on shaft together	EXCLUSIVE
3 Split hub	Easy to mount, even on oversized shaft	ORIGINAL
4 Clamp screw in hub	Holds hub in position on shaft while setting or removing rim	EXCLUSIVE
5 Taper-mated hub and rim	Mount easily in any position—no keyway between taper surfaces	ORIGINAL
6 Large, long pull-up bolts	Hold better and more uniformly	EXCLUSIVE
7 Friction cone grip	Tightening rim on hub gives positive press fit on shaft	ORIGINAL
8 Shaft key lock (set screw over keyway)	Prevents key from drifting	EXCLUSIVE
9 Quickly-detachable (QD)	Pull-up bolts used as jack screws to remove sheave rim	ORIGINAL
10 Interchangeable rims	Hubs for every bore—lower inventory cost	ORIGINAL
11 No realignment problem	Clamped hub holds shaft position—new rim tightens up in exact alignment	EXCLUSIVE
12 Choice of "A", "B", "C" and "D" grooves	"Tailor-made" grooves give proper belt "ride-out" for maximum life efficiency	ORIGINAL
13 I-beam spokes on driven sheave	Stronger—capable of carrying heavier load with less weight	EXCLUSIVE
14 Offset design of sheave	Reduces over-hang loads—increases bearing life	ORIGINAL
15 Worthington-Goodyear V-belts	Load-carrying cords in neutral axis reduce internal friction—each belt carries its share of drive load	ORIGINAL



EASY TO GET ON

EASY TO GET OFF

YET ALWAYS TIGHT ON THE SHAFT

WORTHINGTON QD SHEAVE
—ORIGINAL TAPERED CONE-GRIP SHEAVE

Complete Range of Stock Sizes — Prompt Shipment. 853 listed stock sizes in "A", "B", "C" and "D" sections, fhp to 200 hp . . . 332 listed stock sizes of EC Cord V-belts. *Send coupon for latest Worthington QD Sheave bulletin.*

WORTHINGTON

ORIGINATORS OF THE QD SHEAVE
WORTHINGTON PUMP AND MACHINERY CORPORATION
MULTI-V-DRIVE SALES DIVISION
Buffalo, N. Y. • General Offices, Harrison, New Jersey
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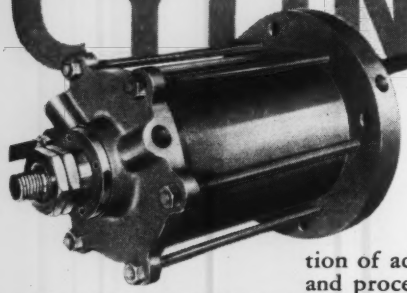
CYLINDERS



Continuous specialization since 1905 in the design and manufacture of cylinders has resulted in Hannifin leadership in the development of improved designs, use of better materials, and introduction of advanced manufacturing methods and processes. Collectively, Hannifin cylinders have chalked up literally hundreds of thousands of service years of experience covering almost every conceivable type of application. You are not experimenting when you specify Hannifin cylinders!

AIR CYLINDERS

- 10 standard bore diameters, 1" to 12".
- Any length stroke. Double acting or single acting.
- 6 basic mounting styles with many double end rod and combination styles.
- Adjustable cushions for head end, rod end, or both.
- Your choice of Series "LW" cup type or Series "R" externally adjustable "Leakproof" piston construction.
- Many special sizes, types available.

HANNIFIN Sets the Standard for QUALITY!

HYDRAULIC CYLINDERS

- 12 standard bore diameters, 1" to 8".
- Any length stroke you specify.
- 11 standard mounting styles, plus many combination styles.
- Standard, double end, or heavy duty (2:1) piston rods.
- Adjustable cushions for head end, rod end, or both.
- Special sizes, types built to order.

There are easier, cheaper ways to build cylinders—*but not to Hannifin's standards for quality!* Every Hannifin cylinder is "TRU-BORED" (specially trued axially, laterally, and for roundness), then honed to a laboratory-controlled internal finish by exclusive Hannifin methods. Hannifin packings and seals are selected only after exhaustive tests. Hannifin designs and Hannifin's precision manufacturing processes insure complete interchangeability of parts. You assure yourself of unquestioned performance when you standardize on Hannifin!

Engineering Recommendations

Let Hannifin engineers help you get the BEST solution for your cylinder problems. See your local factory-trained Hannifin field representative today. Ask for copies of the helpful bulletins illustrated at the right:

Bulletin 110 "HANNIFIN HYDRAULIC CYLINDERS"
Bulletin 210 "HANNIFIN PNEUMATIC CYLINDERS"
SEND FOR YOUR COPIES TODAY!

HANNIFIN CORPORATION

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AIR CYLINDERS • HYDRAULIC CYLINDERS • HYDRAULIC PRESSES
PNEUMATIC PRESSES • HYDRAULIC RIVETERS • AIR CONTROL VALVES

May 19-20—

Institute of the Aeronautical Sciences. Seventh annual personal aircraft meeting to be held at Lassen Hotel, Wichita, Kansas. Robert R. Dexter, 2 East 64th St., New York 21, N. Y., is secretary.

May 25-27—

Society for Experimental Stress Analysis. Spring meeting to be held at the Hotel Statler, Cleveland, Ohio. W. M. Murray, Central Square Station, Cambridge 39, Mass., is secretary-treasurer.

May 29- June 9—

Canadian International Trade Fair to be held at Toronto, Canada. Additional information may be obtained from the International Trade Fair, Exhibition Grounds, Toronto, Canada.

June 1-2—

American Society for Quality Control. Fourth national convention and fifth Midwest conference to be held in Milwaukee, Wis. Additional information is obtainable from Fourth National Convention, Fifth Midwest Conference, P. O. Box 1204, Milwaukee, Wis.

June 4-9—

Society of Automotive Engineers. Summer meeting to be held at French Lick Springs Hotel, French Lick, Ind. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

June 11-16—

American Electroplaters' Society. 37th annual convention to be held at Hotel Statler, Boston, Mass. Fourth International Electrodeposition Conference in collaboration with the Electrodepositors' Technical Society of England to be held concurrently. Additional information is obtainable from American Electroplaters' Society, 473 York Rd., Jenkintown, Pa.

June 22-24—

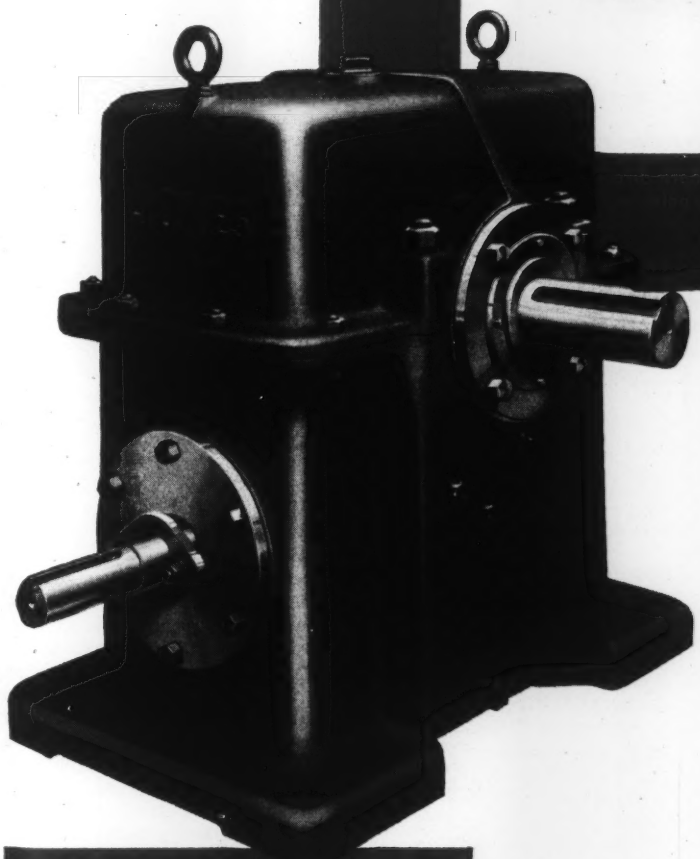
American Society of Mechanical Engineers. Applied mechanics division conference to be held at Purdue University, Lafayette, Ind. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

June 26-30—

American Society for Testing Materials. 53rd annual meeting and ninth exhibit of testing apparatus and related equipment to be held at Chalfonte-Haddon Hall, Atlantic City, N. J. Robert J. Painter, 1916 Race St., Philadelphia 3, Pa., is assistant to the secretary.

Jones

WORM GEAR SPEED REDUCERS



A right angle unit built for heavy duty performance in any industry. The worm can be above or below the gear. Shafts single or double extended for coupling connections or chain or gear drives.

They are built in ratios from 4 to 1 to 90 to 1 and in capacities from ¼ to 100 H.P.

Bulletin No. 68 covers selection tables and dimensions.

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Jones

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PILLOW BLOCKS • FRICTION CLUTCHES • FLEXIBLE COUPLINGS

Simplify Pneumatic Design Problems

with this unique electrically-operated

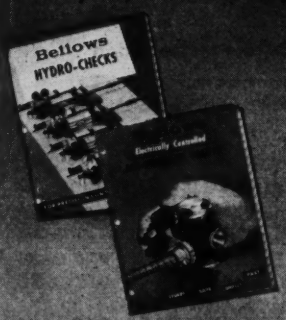
AIR CYLINDER WITH HYDRAULIC CONTROL

MODEL BSM AIR MOTOR

DC-50 HYDRO-CHECK

Permits Positive Control of Piston Speed in Either or both directions and at any point in piston travel.

Provides Absolute smoothness of piston movement — eliminates the natural "bounce" or "springiness" of air.



As a design engineer you'll be interested in knowing more about the Bellows system of pneumatic operation and controls. We'd like to send you two new bulletins showing how "Controlled-Air-Power" operates. No cost. No obligation. Just drop us a note and ask for your copies of Bulletins AV-200 and HC-600. Address The Bellows Co., Dept. MD 450, Akron 9, Ohio.

The Bellows Co.

AKRON, OHIO

DESIGN ABSTRACTS

How An Automobile is Styled

IT IS generally accepted that an industrial designer works best when draped at the edge of a turquoise swimming pool while Nubian slaves in gold sarongs serve chilled nectars in silver cups. Soft music relaxes his nerves while a blonde, streamlined masseuse works on the master's right wrist. His wrist is a little stiff from having endorsed too many checks with a ball-point pen. After being rubbed with pine oil, the industrial designer is ready for the morning chores, consisting of a series of interviews with the nation's foremost magazine writers in search of new material. He then makes ready for the afternoon session which is devoted to press photographers. And so on and on.

Experimental Designs Important

Since you are familiar with all this, let us concentrate on the moments when the master gets fed up with the routine and gets to the design part. If you will bear with me we shall consider the master at work.

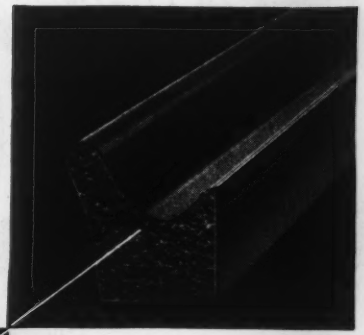
The primary function of an automobile styling division is to develop a design, in close collaboration with the engineering department, which will be as attractive as possible within the strict limitations of cost and other practical considerations. The secondary function of the stylist is to constantly originate other designs of à more advanced nature, free from inhibitions—uninfluenced by immediate criticism from whatever sources. No one, least of all the serious stylist, proposes these experimentals as practical. However, they serve to inspire fresh thinking and to prevent falling into design ruts or grooves. They are intended to stimulate both engineers and designers—to startle them, if necessary, into entirely new channels of design-consciousness.

In order to serve this dual function at the Studebaker plant, an organization is maintained under my direction. My responsibilities towards the management, as head of the styling division, are these: (1) To establish a long-range design philosophy for the company; (2) to develop a practical design for immediate use within the frame of that philosophy; and

An imaginative automotive engineer used

Spongex® cellular rubber

to keep rattles and weather out for greater driving pleasure



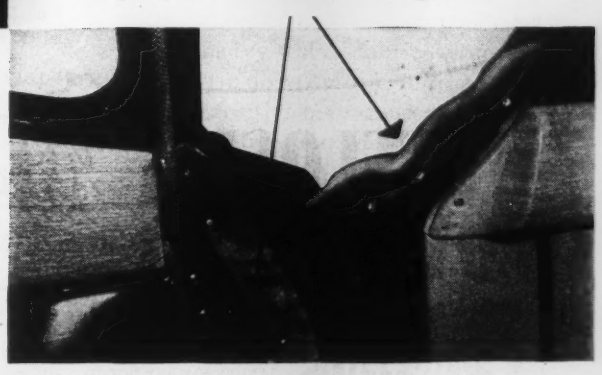
Suppose your car doors shut metal to metal! What a *crash* when you slammed them. What squeaks, rattles and groans as you drove along! And you'd get wet. You would freeze. It would *always* be drafty.

But a sponge rubber product seals out drafts, temperature and moisture . . . cushions shock and vibration . . . absorbs noise. Doors shut snugly and quietly, slam after slam after slam.

If you have a vibration, insulation, cushioning, gasketing, sealing or sound damping problem, think about *Spongex*. Cellular rubber does not become a "product" until you make it one in *your* application. We welcome new problems.

Illustration below—The vertical cord is decorative and functional. This fabric covered Spongex cord seals out drafts.

On door and body are Spongex pieces molded around metal to afford reinforcement, rigidity and mechanical fastening.



Spongex forms shown are from molds designed by us and owned by our customers. As such they are not offered for sale. Similar products can be molded to your requirements in molds designed for you.

TECHNICAL BULLETIN ON SPONGE RUBBER AVAILABLE ON REQUEST

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HY-TEN Steels offer the advantages of the latest metallurgical improvements before they are incorporated in the standard groups. The HY-TEN of today is the standard steel of tomorrow.

HY-TEN is a guarantee of uniform chemistry, grain size, hardenability, etc.

HY-TEN *and* STANDARD AISI and SAE Steels are stocked in a wide variety of sizes, shapes, treatments and finishes, thus assuring prompt reliable steel service from WL's seven warehouses.

Write today for your FREE COPY of the Wheelock, Lovejoy Data Book, indicating your title and company identification. It contains complete technical information on grades, applications, physical properties, tests, heat treating, etc.

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HY-TEN and AISI

BILLETS AND FORGINGS FOR PRODUCTION, TOOL ROOM AND MAINTENANCE REQUIREMENTS.

(3) to maintain a state of alertness and creativeness in the staff.

The styling division is made up of designers, modelers and pattern makers. In order to assist these men in their work, we try to maintain a proper atmosphere or design climate. For instance, we like to have our designers working on several problems simultaneously. Besides the immediate problem—which is always the design of the next production job—we encourage the men to proceed with their advanced designs. By encouraging them to pursue dual programs, a state of mental alertness is sustained that is reflected unconsciously in the design of the production job. Another contributing factor in maintaining the proper design climate is relative freedom in working hours. If the designer is held strictly to conventional business hours, somehow or other his work begins to resemble nine-to-five creation.

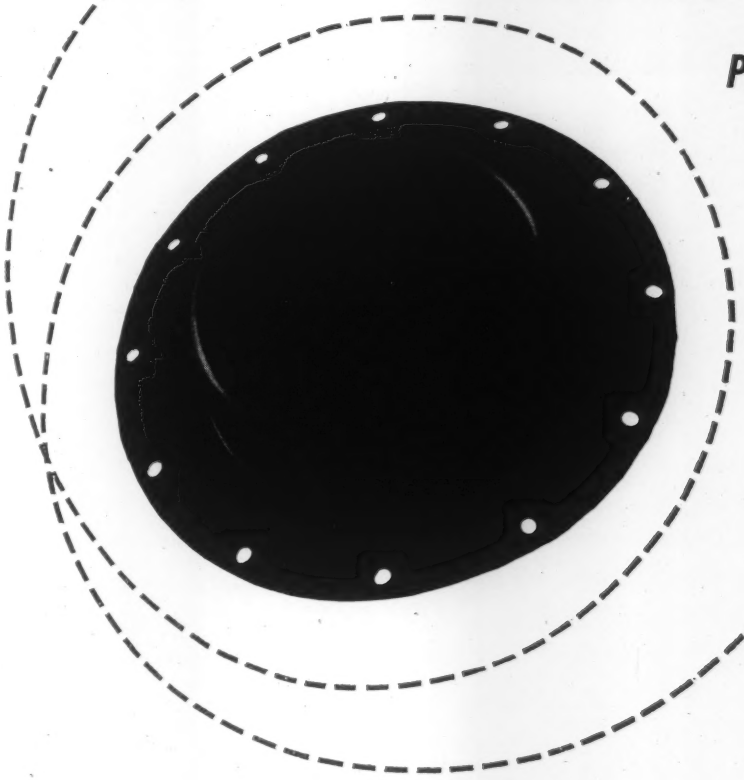
Thirdly, we encourage these designers to get out of the plant, to travel occasionally. South Bend designers are brought to the New York office in rotation where considerable creative activity occurs in widely divergent fields. By giving the men a chance to escape their regular surroundings, to work at a different pace, with different associates, their outlooks are refreshed.

Tactical Strategy

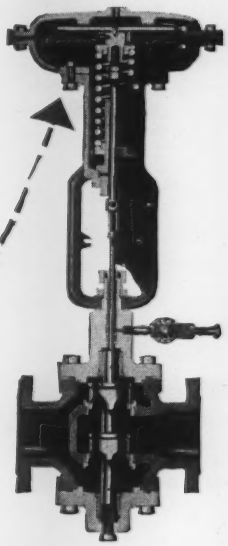
The strategy that my tactical task force is directed to execute in working out a new design is simple. Stylewise I do not believe that keeping in step with the competition is enough. I believe that an independent company in order to succeed, must be courageous and progressive. The results may be somewhat of a shock, but it is far better than blandness. Blandness of design for an independent means pernicious sales anaemia. My staff designers are thoroughly indoctrinated in some fixed principles. The emphasis is always on the same points: (1) Weight must be kept to a minimum; (2) visibility must be excellent; and (3) the automobile must look fast.

A group of designers may be given the task of doing rough sketches of the entire car. Another group will work on front and rear ends only, etc. We establish a time schedule, deadlines, and the work begins. Within a reasonable time we have piles of rough sketches—enough ideas for half a dozen companies—and with the sketches several ⅛-size rough clay models. Some designers prefer to work in clay, others with pencil. We impose no restrictions on the tech-

Keeping the Pulse of Automatic Control . . . with a LINEAR MOULDED DIAPHRAGM



A Special LINEAR Precision Moulded Diaphragm is the heart of this Honeywell Series 700 Control Valve



This high-lift, proportional type valve is precision-engineered from top to bottom. And *just as important* as the flow characteristics of its inner valve are *the sensitivity and dependability of its air-operated diaphragm.*

According to Honeywell engineers, this LINEAR Diaphragm . . . precision moulded to their specifications . . . affords these definite operating advantages:

. . . lasts for millions of cycles.

. . . gives uniform performance under varying temperature conditions.

. . . remains sensitive throughout its service life.

. . . withstands accidental overload pressures.

. . . permits smooth action for minute movements.

. . . provides a pressure-tight seal without strain on the cast flange.

Your diaphragm problems can be solved, by LINEAR, with the same special engineering attention. Whether you need heavy-duty, fabric-reinforced diaphragms or the super-sensitive homogeneous type . . . Your individual specifications will be our guide. Write, wire or call LINEAR today!

"PERFECTLY ENGINEERED PACKINGS"

LINEAR

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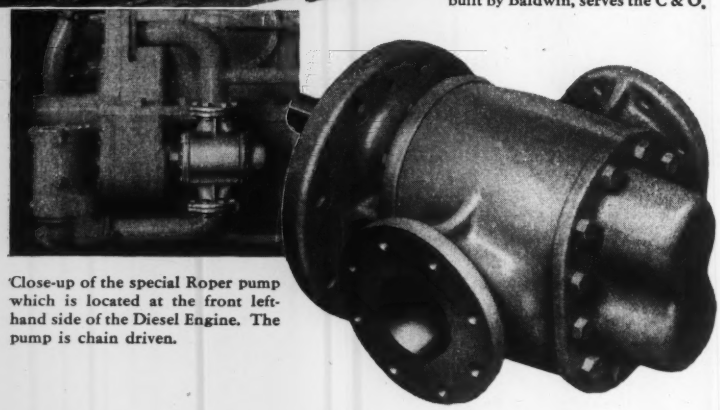
ROPER Rotary PUMPS

DELIVER LUBRICATING OIL ON BALDWIN ROAD-SWITCHING LOCOMOTIVES...150 G.P.M. at 100 P.S.I.

This 1500 HP Diesel Locomotive, built by Baldwin, serves the C & O.



Close-up of the special Roper pump which is located at the front left-hand side of the Diesel Engine. The pump is chain driven.

A TYPICAL "CUSTOM-BUILT" APPLICATION BY ROPER

This custom-built pump — made to deliver lubricating oil on a Baldwin Diesel Switching Locomotive — is typical of a wide range of tailored-to-the-job units produced by Roper. Although applications may vary greatly, each custom pump is designed along the basic Roper principle employing only two moving parts. The simplicity of this design — equal size, helical pumping gears operating in a case with proper clearance — promotes high efficiency and long service life. Roper custom-built — as well as Roper standard models — can effect substantial savings in the long run. Sizes range from 3/4 to 300 G.P.M. — pressures up to 1000 P.S.I.

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244 Blackhawk Park Ave.
ROCKFORD, ILLINOIS

ROPER Rotary Pumps

DEPENDABILITY SINCE 1857

niques they employ.

Now the important process of elimination begins. From the roughs, I select the designs which indicate germinal direction. Those that show the greatest promise are studied in detail and these in turn are used in combination or arrangements with one another. A promising front treatment can be tried in combination with a likely side elevation sketch, etc. From this a new set of designs emerges. These, then are sketched in detail. After careful analysis they boil down to four or five. Of these, quarter-scale clay models are started. And again, I want to remind you, these same designers are working on any advanced idea which intrigues them and they may be working out this blue sky stuff in ⅛-scale clay roughs or in sketch form.

When models are satisfactory we cast them in plaster, paint them, trim them with bumpers, door handles, lucite headlights, etc. They look like enticing small completed automobiles. They are all practical and within clearance and manufacturing limitations as well.

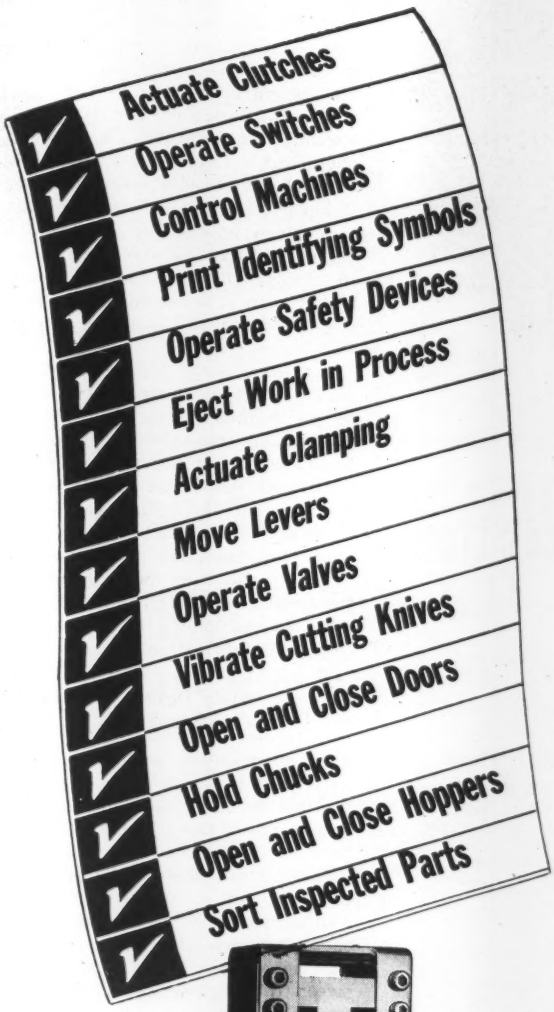
Now we are ready to select the best designs for full-size mock-ups. When it is decided what model to build, a full-scale automobile is made in clay, a method which permits correction of the distortions which invariably occur in the process of "scaling-up" from ¼-inch size. After these corrections are made, a full size plaster cast or a wood mock-up is made.

Now comes the time for the management to make the decision. Changes are suggested inevitably—and another complete showing is arranged to demonstrate how these alterations have been incorporated in the design. When the final approval for production is given the design cycle is complete. It is up to the engineering and production dpartments to draft it and detail it. *From a paper by Raymond Loewy, Raymond Loewy Associates, at the Jan.* 1950 *SAE Annual Meeting in Detroit.*

Reducing Production Costs

GENERALLY there are two spheres of operation in which the problem of reducing costs can be most readily attacked: The design of the product and the fabrication methods. So far as the design of a product is concerned, the first requisite is that the design is such that the product properly fulfills the function for which it is intended.

Once this basic requirement is taken care of, the designer can turn his attention to considering the man-



Namco Push-Type Solenoid with Saddle Stop

Namco "Stellite"—welded Solenoids are built in sizes with ratings from 2½ to 25 pounds, push or pull, at 1" stroke. Combination push-pull and other special applications are available.

If you have jobs like these... then you need SOLENOIDS

They're the modern, up-to-date way of doing jobs automatically—by remote control, and in cramped quarters if necessary.

With the positive, accurate action of Solenoids you can simplify design, manufacture and operation. They're taking the place of expensive, bulky gear trains, levers, linkages and other mechanisms. Many a manufacturer has come to us with a cost or space problem—and has found the solution in Namco Solenoids.

Namco Solenoids—with "Stellite"—welded contacts, are compact, rugged and reliable. They're *engineered* to the requirements of the job, with the aid of an *expert*. That's where we can help you. We'll be glad to recommend the size and style best suited to your job—with standard or special terminal blocks and mountings. Like more details? Ask for bulletin EM-46A.

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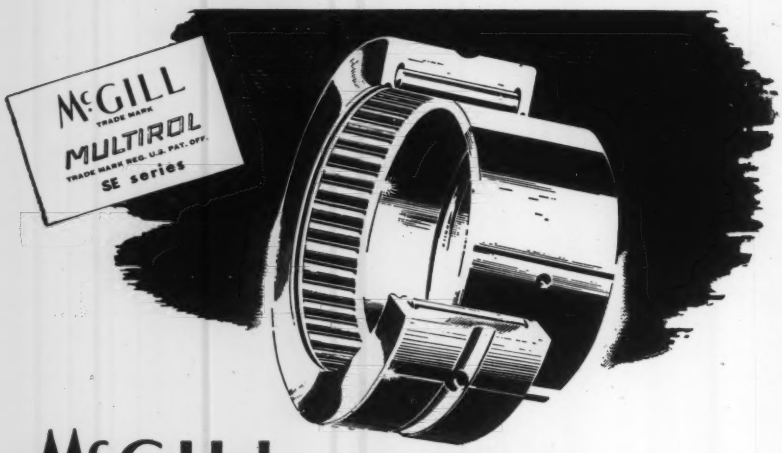
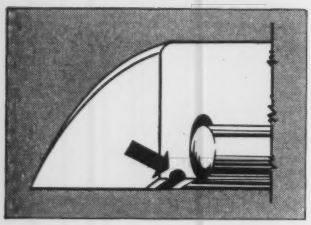
Acme-Gridley Bar and Chucking Automatics: 1-4-6 and 8 Spindle • Hydraulic Thread Rolling Machines • Automatic Threading Dies and Taps • The Chronolog-Limit, Motor Starter and Control Station Switches • Solenoids Centrifuges • Contract Manufacturing



McGILL design provides EXTRA oil seal protection



Extra protection against destructive foreign particles is provided by this lubricant-retaining groove, built into the inside diameter surface of the roller retaining-end shoulder. The ring of lubricant, held in this groove, acts against the inner race O.D., increasing the sealing effect of the precision tolerances between these shoulders and the inner race. Notice, too, the groove above the rounded end of the roller. Acting as a lubricant reservoir, this provides Multirol SE Series bearings with an extra margin of protection against neglected or delayed lubrication.

The increased life and lowered maintenance costs of Multirol bearings are accounted for by these built-in features, plus such design factors as greater load capacity and the elimination of loose or welded-in retaining rings. Multirol SE Series bearings, used with or without inner raceways in standard sizes of 5/8" to 9¼", are used extensively in machine tools, pumps, polishing machines, presses, power shovels, etc. — giving users a high degree of precision performance and longer bearing life.

Why not consult McGill engineers for helpful recommendations in applying these benefits to your product? Write McGill Mfg. Co., Inc., Bearing Division, 200 N. Campbell St., Valparaiso, Indiana. Ask for Bulletin SE-47.

McGILL® precision bearings

ner in which the product can be most economically produced. It is, therefore, essential that the design engineer be thoroughly familiar with all the various types of production equipment which his company has in operation, or available for use. He should also have a working knowledge of the relative costs of various production processes. For example, if a large number of parts are involved it may be more economical to design the part as a pressing rather than as a casting, which might require several expensive machine tool operations.

However, it is hardly to be expected that the design engineer will be as familiar with the details of production equipment, methods and processes, as is necessary to insure maximum economies so far as production costs are concerned. Hence, it is advisable that he work closely with the "methods and process engineer," who may be a member of the general engineering staff or who may come under the manager in charge of fabrication.

Changes While Design is in Progress

In order to properly execute his job, this methods and process engineer must have a wealth of practical fabricating experience and also some knowledge of the principles of design, because it is his responsibility to make certain that each part is so designed that it can be fabricated at a minimum cost using the existing shop facilities. In this part of his duties he is acting as the link between the engineering and manufacturing departments. It is the duty of the methods and process engineer to suggest modifications in design *as the design is in progress* and not to wait until the design is completed. If this is done, a great deal of time and expense can be saved.

While the methods and process engineer has an important part to play in the introduction of new designs originating in the design department, probably his most important function is in initiating modifications in existing designs, which allow cost reductions to be effected by improved production methods and processes. He will also direct changes in production methods and processes to secure cost reductions which may not involve any changes in design. When this can be done, difficulties connected with the interchangeability of parts on old and new models of the product can be avoided.

These two spheres, namely, the design of the product and the methods used to produce the product, are the most obvious ones to be given attention in any cost reduction pro-

DIRECT-TO-YOU SHIPMENTS OF BRONZE BAR STOCK CUT BEARING COSTS!

Stocks of N-B-M "Tiger" Bronze now located for fast delivery of the bar size you want, when you want it.

National Bearing Division has announced a new direct-shipment policy on "Tiger" Bronze Bar Stock—a policy designed to substantially reduce costs of bearings, *used for plant maintenance or on production lines.*

Stocks have been conveniently "spotted" to assure fast delivery. All popular sizes in 13" lengths are available—as-cast or machined cored and solids, or as-cast hexagons.

Product Designers and Plant Engineers with an eye on bearing costs are urged to get the complete facts.

Inquiries receive prompt attention.

"TIGER" BRONZE . . . the ONE Bronze Alloy with ALL these features for longer bearing service at lower cost . . .

WEAR-RESISTANT—Has correct balance between bronze matrix and lead. *Lasts longer.*

ANTI-FRICTIONAL—Low coefficient of friction helps prevent shaft seizure. *Saves power.*

SHOCK-RESISTANT, EMBEDDABLE—Hard enough to stand up under heavy bearing loads. Soft enough to embed foreign material. *Protects the shaft.*

EASY TO MACHINE—With speeds as high as 3000 F.P.M. *Saves time.*

USE THIS COUPON —*for price quotations by return mail, and new bulletin giving complete FACTS on physical properties, operating characteristics, bar types and sizes.*

Please send me your new Cored and Solid Bar Bulletin and prices on "Tiger" Bronze:

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We use Bronze Stock for .....

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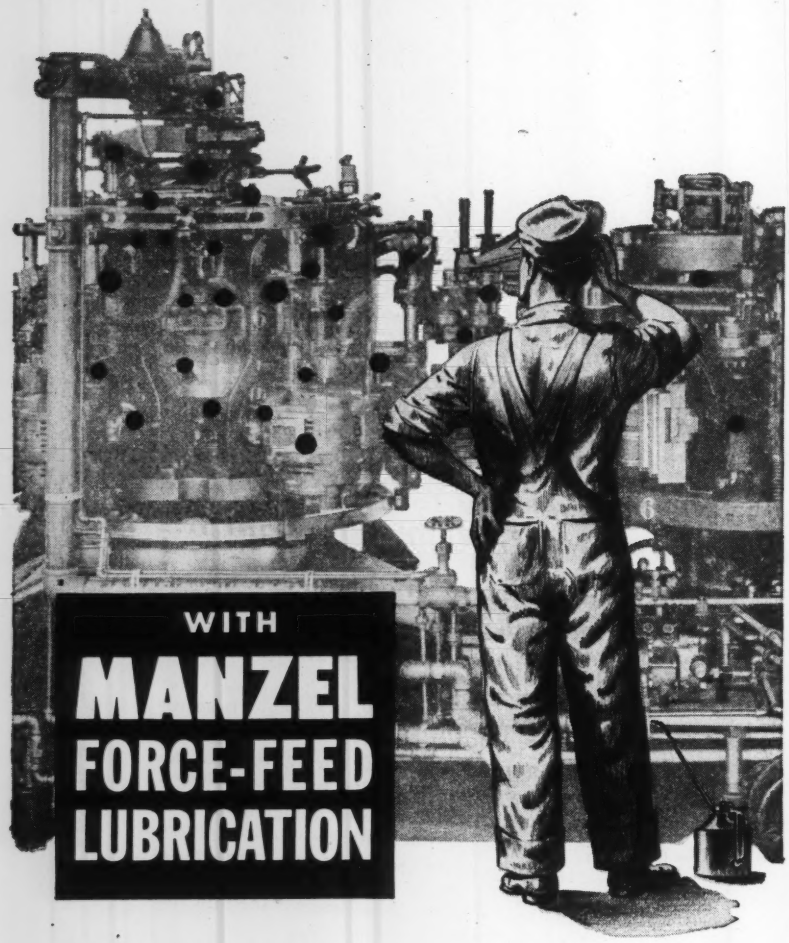
CORRECT THESE SPOTS BEFORE HIS EYES!



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MANZEL FORCE-FEED LUBRICATION

MANY spots requiring many different amounts of lubricant make it easy for a new man (or even an experienced one) to get all mixed up.

Manzel Force Feed Lubricators eliminate not only the confusion—but also the workman. For Manzels automatically lubricate any number of wearing points—no matter how hard to get at. Just enough of the right type of lubricant is always at each point.

Manzel Force Feed Lubricators save the initial cost many times over each year. They are supplied as standard equipment on various makes of engines, pumps, compressors, and other heavy machinery. Or they can be installed on your present equipment.

A Manzel representative will gladly give you technical assistance on your lubrication problem.

Write us now.

Manzel Inc. now supplies repair parts for all models of Bowser and Torrington Lubricators.

Builders of HIGH PRESSURE METERING PUMPS Since 1898

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Manzel INC.

gram. There are, however, many instances in which additional cost reductions may be achieved by the careful selection of raw materials and semifinished or finished parts which are purchased to form part of the final product. *From a paper by A. F. Davis, vice president, Lincoln Electric Co., presented at a meeting of the Cleveland Engineering Society, Feb.* 1, 1950.

Aircraft Crashworthiness

IN OUR efforts toward the achievement of aviation safety, we have quite properly been concerned primarily with the prevention of accidents. This has been, and will continue to be, the main objective toward which our efforts should be directed.

Realistically, however, we must recognize the fact that despite our best efforts the ideal of complete elimination of accidents is not attainable. Completely accident-free operation could be achieved only by infallible man, but since all personnel concerned with aviation—whether they be designers, factory personnel, operators, maintenance mechanics, or pilots—are human, and subject to human error, we must face the inevitability of accidents.

Therefore, the problem of "crashworthiness," of assuring the highest practicable survival rate, is as much a part of our responsibility as is the prevention of accidents. The term crashworthiness does not mean and must not be considered to imply crashproof airplanes. It *does* mean providing the maximum practicable degree of occupant protection in those crashes in which aircraft damage is such that the accident may be considered "survivable."

This problem of crashworthiness applies to all aircraft, whether they be personal, transport or military, although the degree of protection which may practicably be provided may vary for the different categories.

Reducing Fatalities One Fourth

The total number of passenger fatalities in all accidents for the eleven year period from January 1, 1938, through December 31, 1948 was 672, of which 169 occurred in survivable accidents. Therefore, if it be assumed that adequate crashworthiness could have effected 100 per cent passenger survival in the survivable accidents, there would have been a reduction of 25 per cent in the total number of fatalities.

The entire problem may be summed

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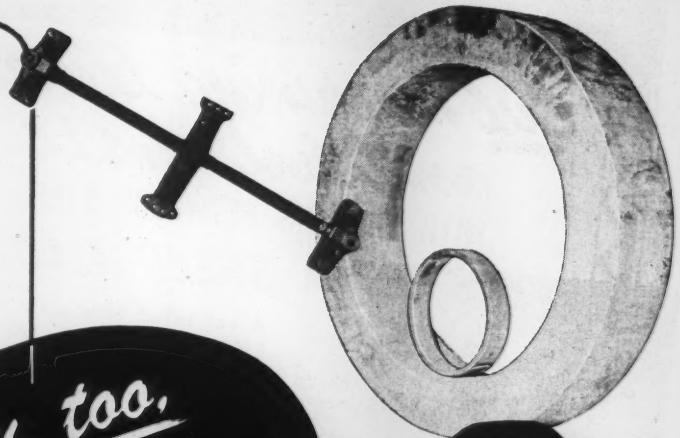

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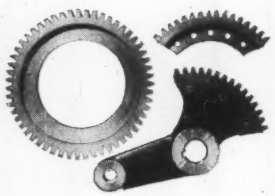
The chromium and molybdenum content of SPEED ALLOY qualify it for tough jobs like this sector gear made by United Precision Gear Co., Indianapolis, Ind. After trepanning the center and cutting the teeth into the outside diameter, the circle is cut into segments and heat treated. No distortion at all was experienced. Finished parts were fastened to the hitch feeding mechanism of a large punch press — Undoubtedly SPEED ALLOY can cut your costs and solve your tough jobs—plates stocked up to 72" wide and 6" thick flame cut to specifications and specially processed before shipment to insure maximum machinability—Send for bulletin #905.



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up by the statement, "A reorientation of the attitude of the aircraft designer and abandonment of the polite fiction that transport aircraft do not have accidents are overdue. Accidents are still occurring and there is nothing to indicate that they will not continue to occur in the foreseeable future. It is reasonable that aircraft should meet crashworthiness as well as airworthiness requirements." The inadequacy of the presently required level of crashworthiness is clearly established by the record of needless loss of life in survivable accidents.

An analysis of load factors developed in survivable accidents reveals that decelerations up to 20*g* are extremely common, and that higher values are encountered. Therefore, 20*g* protection represents the minimum acceptable level of crashworthiness, and higher levels up to 40*g* are desirable and should be incorporated wherever possible. Protection for sideward and upward acting forces should be increased correspondingly. Such levels of protection are needed in all types of airplanes—transport, personal and military. The upper level of human tolerance to crash decelerations has not been determined, but it has been shown to be far in excess of the recommended level of protection.

An analysis of the weight penalty involved in providing protection of the order indicated shows that it need not be high, and is less than that involved in other emergency provisions. The effect of a reduction in fatality rates on public acceptance of flying can more than offset any economic loss entailed in the increase in weight. *From a paper by William I. Stieglitz, design engineer, Republic Aviation Corp., presented at a recent meeting of Institute of the Aeronautical Sciences Inc. in New York.*

Uncharging Plastics

STATIC electricity has for years created serious difficulties in the plastics, textile, and paper industries. It engenders many problems of the annoyance class in a wide range of fields and activities. Any place where nonmetallic materials must be handled in quantity or with speed and where electrical nonconductors are used is fair game for static electricity.

One process for static elimination, known as Electrosol, consists of a two-stage dip or spray application of two successive formulations. The first material is a fluid consisting essentially of an electrical insulator, a dispersing and wetting agent, a cata-

field-tested and proven...

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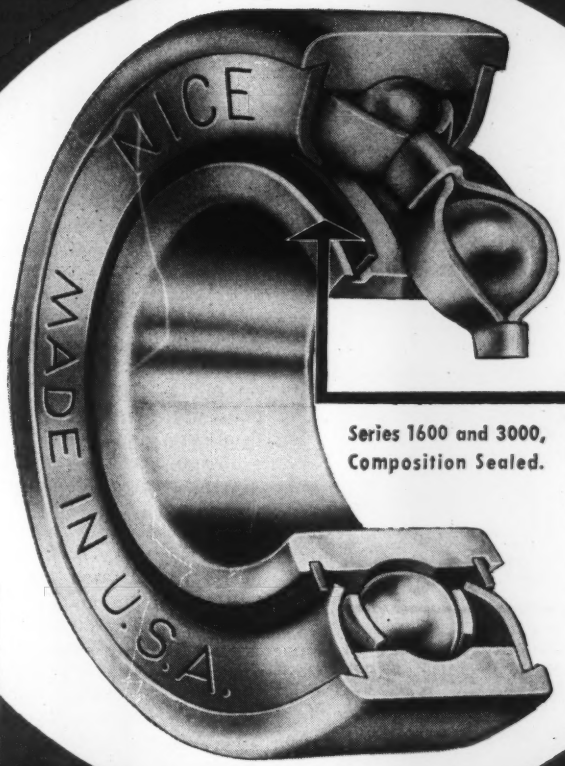

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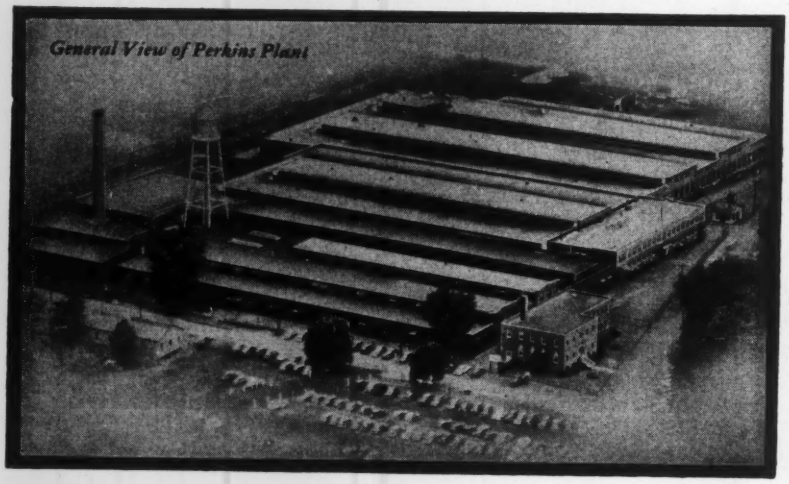

lyst for the complete process reaction and a vehicle to provide uniform deposit, control viscosity and permit uniform pull-out in dip usage.

After a dip or spray application and a twelve minute air dry at normal room temperature, the plastic is given an application of another fluid. The result of the second application is an unchanged original material, the surface of which can not be electrostatically oriented and which prevents orientation of the parent material it covers. Hence, no static charge can exist. The total deposit on the parts or material processed averages 0.00035-inch in thickness.

Interesting applications have suggested themselves for investigation and use. Transparent plastic molds of operating machinery, selling aids, and plastic inspection ports in machinery are obscured by lint and foreign material due to the static charge generated by the rotating equipment within. Such difficulties are readily cured with this treatment.

Molded windows and bubbles for automotive and aircraft use experience substantial accumulations of static electricity because of the rush of air past them. Resultant attraction of foreign matter requiring excessive cleaning entails loss of clarity. Static build-up and electrical discharge from such areas can disturb operation of radios, communications and other electronic equipment. Processing of the parts after molding or forming effectively relieves this.

Filtering Can Be Dangerous

No discussion of static electricity would be complete without a word in connection with a little known phenomena which takes its yearly toll in the plastics industry and elsewhere. In work with plastics where volumes of inflammable solvents are used, such solvents frequently become undesirably contaminated or dirty. The seemingly logical, and frequently disastrous, thing to do is to filter such materials through filters, chamois, paper, ceramic, or other porous material. When such action is performed, a very marked separation occurs not only of the dirt from the solvent, but of the positive from the negative charges within the solvent. A spark originates and another unexplained fire or explosion results. A review of solvent handling operations with this in mind, particularly those not performed completely in closed systems with all oxygen excluded, is certainly time well spent. *From a paper by Merrill J. Ainsworth, engineering consultant, at the Jan., 1950 meeting of the Society of Plastics Engineers at Cleveland.*

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YOU CAN BE **SURE**.. IF IT'S

Westinghouse

Life-Line's Space-miser compactness SAVES PRODUCTION COSTS

This study in contrast shows how a Life-Line comes through in a pinch. For example: One manufacturer using conventional motors was forced to crowd two motors together. They did not quite fit, which meant removing a thin slice of cast iron from each motor frame. He'd been doing it for years . . . thought nothing of it.

Then, a designer saw the all-steel Life-Line. He tried it. Result? A step saved in manufacturing. Life-Line's ⅓ smaller size, for the same frame size and hp rating, fit easily with room to spare.

This compactness is a valuable asset when your design allows only limited space for the motor drive. (And saves money in reduction of shipping weight, too.) In addition, Life-Line "needs no further lubrication". You can install it in "hard-to-get-at" locations—and forget it!

These are but a few of the pluses a designer has with the Life-Line. Ask your Westinghouse representative for others. Ask him for a copy of B-3482, or write Westinghouse Electric Corporation, P.O. Box 868, Pittsburgh 30, Pennsylvania.

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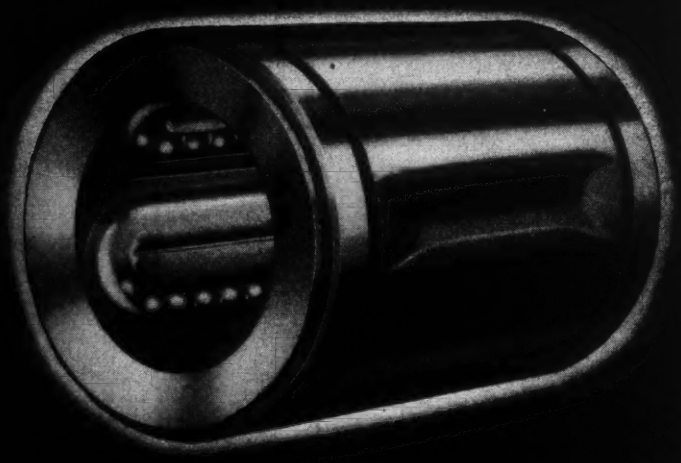

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Sliding linear motions are nearly always troublesome. Thousands of progressive engineers have solved this problem by application of the Precision Series A Ball Bushing.

The low-cost Commercial Grade Series B bearing is now added to the Ball Bushing line and offered to original equipment manufacturers. This ball bearing has been developed for support of linear motions in competitively priced, volume produced products where super precision is not essential. Alert designers can now make tremendous improvements in their products by using Ball Bushings on guide rods, reciprocating shafts, push-pull actions, or for support of any mechanism that is moved or shifted in a straight line.

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Now in production for ¼", ⅜", ¾", 1" and 1½" shaft diameters. If you have a catalog, phone your representative or write us for new Data Sheet. If not, write for complete literature and the name of our representative in your city.

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ACCOUNTING MACHINE. Portable, high-speed model for automobile dealers. For posting daily journals, accounts receivable, accounts payable, general ledger, payroll and other records. Machine also adds, subtracts, multiplies and divides. Has 10-key natural sequence keyboard for touch fingering. Underwood Corp., New York, N. Y.

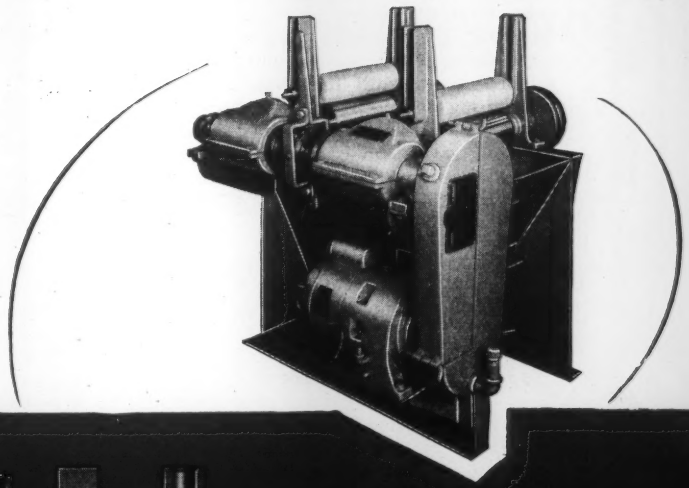
ELECTRIC TYPEWRITER. Improvements include 1-space tabulation, governor-controlled tab-key carriage return for wide tabulations, faster space bar operation and key resistance reduced 50 per cent to 2½ oz. In gray wrinkle finish with black plastic keys having injection molded characters. Remington Rand Inc., New York, N. Y.

ADDING TABULATOR. Actuated automatically or manually to print in vertical or horizontal sequence. Carriage and automatic controlling mechanism tabulate and return in unison or operate independently. For railroads, banks, post offices, insurance companies, etc. Have direct subtraction and credit balance with list capacity of 10 columns and totaling capacity of 11 columns. Prints items in black with totals and sub-totals in red. Underwood Corp., New York, N. Y.

ELECTRIC TYPEWRITER. Keyboard positions identical with standard and portable typewriters. Electrically-controlled touch adjustable to individual's typing touch. Key lever dip uniform over entire keyboard. Margins set automatically by touching button. Repeat underscore eliminates constant tapping of underscore key when underlining words. Royal Typewriter Co. Inc., New York, N. Y.

ELECTRIC TYPEWRITER. Keyboard-controlled electric margins enable operator to set left and right margins without raising hands from keyboard. Rubber insulation between operating mechanism and machine provide internal cushioning to isolate and reduce noise transmission and vibration. Underwood Corp., New York, N. Y.

MECHANICAL ACCOUNTING MACHINE. Features automatic control of its operations through sensing unit attached to carriage. Price and per-



For nearly a half century, many power transmission problems have been presented to Carlyle Johnson engineers by builders of textile machinery. The Maxitorq floating disc Clutch seems to be well suited to this specific field . . . as is evidenced by the installation shown above.

The H. W. Butterworth & Sons Co., has selected a Maxitorq No. 22 double type as original equipment in their newest No. 657 Laboratory Jigger.

Separator Springs keep "floating discs" free in neutral . . . consequently there's no drag, no abrasion, no heating. They also assure instant, complete disengagement. Assembly, adjustment, and take-apart are manual. Clutches are shipped completely assembled, ready to slip onto a shaft.

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formance between small portable bookkeeping machines and highly specialized automatic accounting equipment. Four control units included to permit handling four different accounting jobs. New carriage movement permits machine to print while tabulating in either direction. Burroughs Adding Machine Co., Detroit, Mich.

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ELECTRONIC AIR CLEANER. Precipitron unit restyled in smaller cabinet. Air flow changed to horizontal straight-through system. Power requirements reduced 25 per cent to 60 watts; unit cleans 1200 to 1500 cfm air. Access to interior through top panel. Includes addition of 2 water spray nozzles and new control timer that automatically sequences washing cycle once initiated. Westinghouse Electric Corp., Boston, Mass.

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INDUCTION HEATER. Smith-Dolan system unit designed so each half is energized independently by own contactor. Voltage adjustments made by rotary switch which cannot be changed when heater is carrying load. Both halves can be controlled as one integrated unit. Machine is mobile and self-contained, with 60 and 75-kva fan-cooled units. Electric-Arc Inc., Newark, N. J.

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SHEAR. For cutting irregular shapes in mild or stainless up to $\frac{3}{16}$-in. thick. Attachment for circle and strip cutting provided; shear will also cut to scribed line or template. Exchanging cutting tools for hammer type dies permits use for beading and forming in steel to $\frac{1}{8}$-in. thick. Shear has 48-in. throat, variable stroke adjustment, and is powered by 1½-hp totally-enclosed 220/440-volt, 3-phase, 60-cycle motor. Baker Bros. Inc., Toledo, O.

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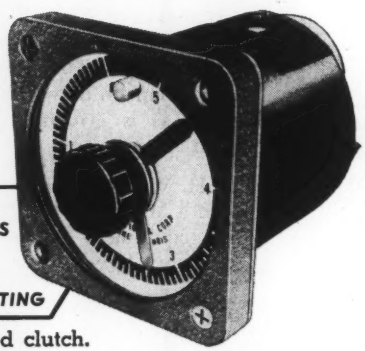

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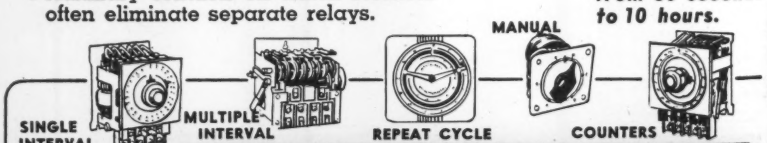

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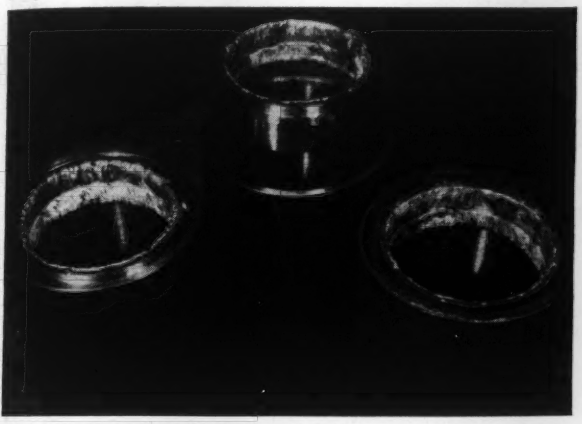

Fig. 1. Production up 300%. *Above (center) is honed cylinder ready for steel insert. At left, assembly is ready for "Manual Lincolnwelding." At right, solid one-piece unit after welding.*

Fig. 2. "Manual Lincolnwelding" *at The Ohio Honing and Hydraulic Co., Cleveland, Ohio. Fixture rotates past welding gun. Note absence of spatter and smoke for easier, safer welding.*

By ALBERT BLEWETT, President
The Ohio Honing and Hydraulic Co.
Cleveland, Ohio

Dense, uniform welds now being produced faster and easier with the "Manual Lincolnweld" are eliminating former difficulties of leakages in hydraulic cylinders resulting from porosity in welds. Hard spots in weld metal have been overcome, greatly simplifying subsequent machining operations. In addition to a better design, the rate of production has been increased 300%.

The cylinder is made from high carbon tubing (Fig. 1) that has been previously bored and honed. The 1020 steel insert, faced on the inside diameter with bronze, is pressed and welded to the cylinder. The welding is done in a rotary fixture (Fig. 2) that moves the work past the electrode. With "Manual Lincolnweld" the electrode is fed automatically into the work at a predetermined rate set on the welder controls. The granular flux, fed from the welding "gun", maintains a protective blanket over the arc to shield the arc during welding and thus produces a smooth, uniform weld having deep penetration.

A flux dam (Fig. 3) made from a carbon ring is used to confine the granular flux and facilitate easy recovery of unused flux with a vacuum recovery unit. The slag is then quickly removed with an air operated chipper. After welding, the end of the cylinder is faced and the insert bored. The final operation is simply to grind the outside diameter of the cylinder to size.

WELDING IS DONE AFTER MACHINING

With "Manual Lincolnweld," the welding can be done after the inside diameter of the cylinder has been honed to size. By means of its concentrated arc welding beneath the covering of granular flux, welds can be completed with negligible distortion.

Fig. 3. Removes Flux *with vacuum recovery unit shown above, at left. At right is chipping of weld slag in trough of flux dam.*

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HAND SHEAR. No. 206 Unishear cuts 6 gage steel at speeds to 5 fpm. Specially designed for straight cut, with blade motion feeding in the work. Universal motor operates on 115, 230 or 250-volt dc, or 60 cycle or less ac. Net weight, 38 lb. Stanley Electric Tools, New Britain, Conn.

BENCH GRINDER. For light grinding, buffing, polishing and wirebrush work. Has adjustable tool rests and steel wheel guards that are adjustable to permit grinding at any point on wheel circumference. Powered by ¼-hp, 3450-rpm motor. Net weight, 34 lb. Stanley Electric Tools, New Britain, Conn.

DRILLING AND TAPPING HEAD. Multiple head available with 2, 3, 4, or 5 spindles. Drilling and tapping capacity, $\frac{5}{16}$-in. in steel. Center distances adjustable from $1\frac{1}{16}$ to $7\frac{5}{8}$ in. Spindles can be located at any point within $3\frac{5}{16}$-in. diameter circle. Bed plate ties components together like punch and die. Ettco Tool Co. Inc., Brooklyn, N. Y.

BENCH TAPPER. Torquematic unit for bench tapping and drilling operations. In 3 models; 0 to $\frac{3}{16}$-in.; $\frac{3}{16}$ to $\frac{5}{16}$-in.; and $\frac{5}{16}$ to $\frac{5}{8}$-in. Unit belt driven from desired motor with belt covered for operator safety. Charles L. Jarvis Co., Middletown, Conn.

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TAPPING HEAD. For large hole tapping. Capacity, ⅜ to 1 in. in steel. Features smooth action friction clutch, spline drive to tap holder and helical back gear reversing mechanism, with # 3, # 4 or #5 Morse taper shank. Procunier Safety Chuck Co., Chicago, Ill.

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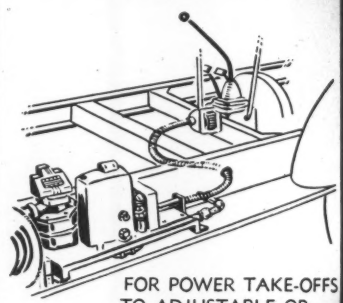

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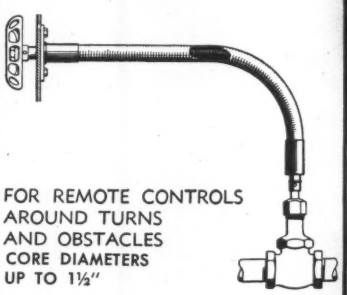

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CORE DIAMETERS UP TO 1½"

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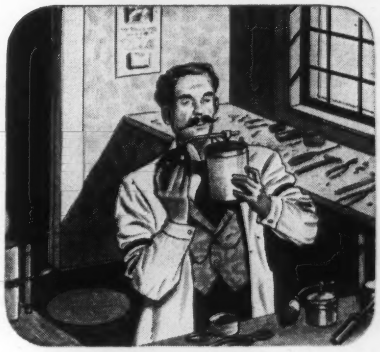
Backed by 50 years of experience in the Flexible Shaft and related fields, the Elliott engineering staff is constantly incorporating new and modern improvements in the design of their own equipment. This staff is available to you, as a part of the Elliott Service, to cooperate in working out the details of your own problems. We will be pleased to supply Flexible Shaft Data Sheets on request. These will aid you in furnishing us with certain necessary specifications such as r.p.m., torque, deflection, etc.

Write for Data Sheets and for Catalog 49 on your Company Letterhead

ELLIOTT
MANUFACTURING COMPANY
217 PROSPECT AVE. BINGHAMTON, N. Y.

1 1875—For centuries, hard labor and a paintbrush were man's main means of adding beauty and color. But, with the industrial revolution at hand, new devices were sorely needed to break this bottleneck.

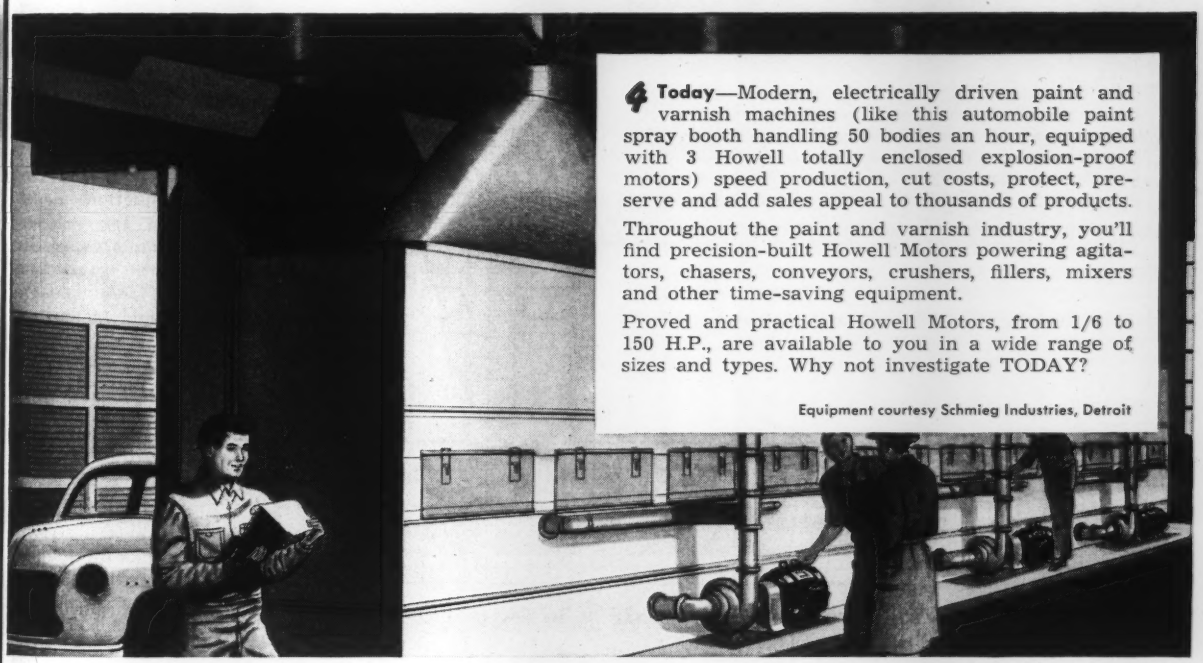


2 1900—The artist's airbrush was born. On its heels came the paint spray gun. Both products, strangely enough, stemmed from the atomizer used by doctors to prevent colds. Painting swung into high gear.

3 1915—Portable paint-spraying equipment was introduced in this, the year Howell "Red Band" Electric Motors arrived. Soon these rugged motors made important contributions to this and other industries.

PAINT . . . protects, preserves, decorates and sells!

(AND HOWELL MOTORS HELP SPEED MODERN PAINTING PROCESSES—CUT COSTS)



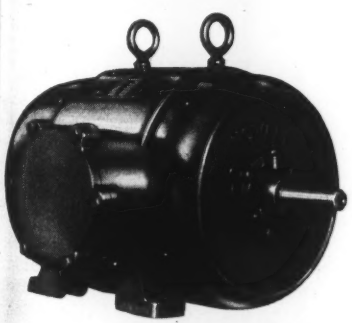
4 Today—Modern, electrically driven paint and varnish machines (like this automobile paint spray booth handling 50 bodies an hour, equipped with 3 Howell totally enclosed explosion-proof motors) speed production, cut costs, protect, preserve and add sales appeal to thousands of products.

Throughout the paint and varnish industry, you'll find precision-built Howell Motors powering agitators, chasers, conveyors, crushers, fillers, mixers and other time-saving equipment.

Proved and practical Howell Motors, from 1/6 to 150 H.P., are available to you in a wide range of sizes and types. Why not investigate TODAY?

Equipment courtesy Schmieg Industries, Detroit

Free enterprise encourages mass production, supplies more jobs—provides more goods for more people at less cost.



Howell totally-enclosed, explosion-proof motors for use where inflammable liquids or gases are handled or stored.

HOWELL MOTORS

HOWELL ELECTRIC MOTORS CO., HOWELL, MICH.

Precision-built Industrial Motors Since 1915

HOWELL Red Band MOTORS

FRACTIONAL... HORSE POWER GEARS

PRECISION WITH PRODUCTION VOLUME

Internal
Helical
Spur
Elliptical
Worm Gear
Worms
Ratchets
Sprockets

Consult our Gear Experts on your...next problem

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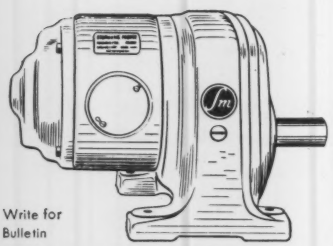
Costs Going Up?

Pull them down with Sterling Slo-Speed electric power drives.



Write for Bulletin

STERLING ELECTRIC MOTORS

Plants: New York • Los Angeles • Hamilton, Canada
Offices in Principal Cities

SLO-SPEED (Geared) • SPEED-TROL (Variable Speed) • KLOSD (Normal Speed)

ing vibration. Equipped with wheel guards and tool rests, with exhaust connections available. The Hisey-Wolf Machine Co., Cincinnati, O.

TOOL GRINDER. No. 10N cutter and tool grinding machine for sharpening of plain milling cutters, formed cutters, saddle and face mills, side milling cutters, end mills, etc., and for light cylindrical grinding, internal and surface grinding. Has operating controls at front and rear of machine. Double-ended wheel spindle swivels in horizontal plane. Transverse movement screw mounted over V-way and ground steel rollers eliminate carriage twist. Brown & Sharpe Mfg. Co., Providence, R. I.

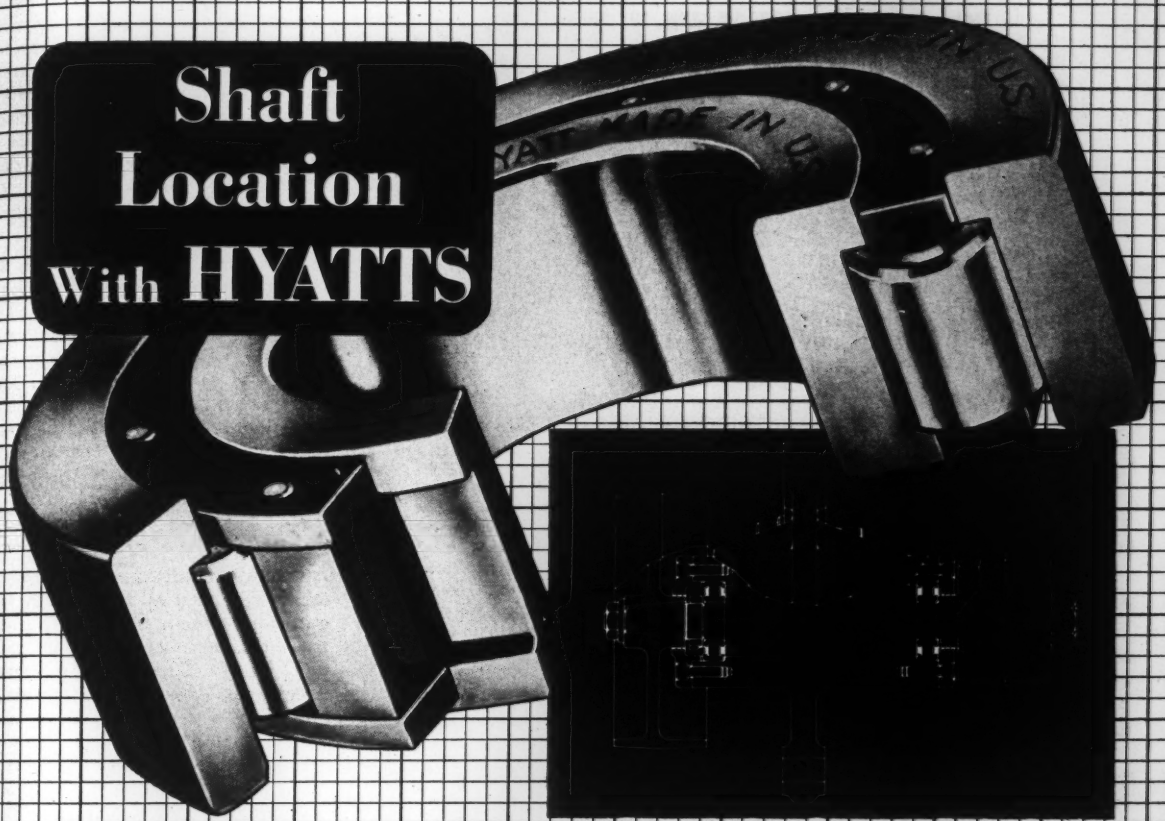
DIECASTING MACHINE. Capacity, 1 lb zinc alloy. Utilizes fast chill and high cycling speed on low cost, single impression dies. Direct valving injects air into cylinder, forcing molten metal into dies at high speed before cavity has cooled. Average production of 500-700 shots per hour possible with die sizes to 6 by 9 in. Economical for production runs as low as 2000 pieces. DCMT Die Casting Machine Corp., New York, N. Y.

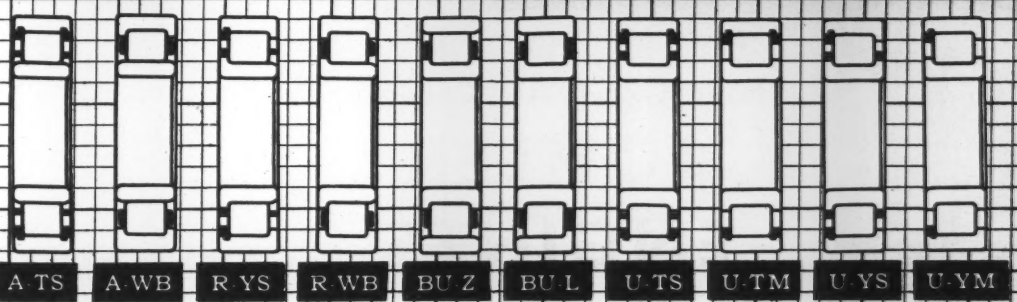
HONING MACHINE. General purpose, heavy-duty semiproduction model. Particularly suitable for reconditioning 5 to 8 automotive blocks per hour or 60 to 90 connecting rods per hour with from 0.004 to 0.01-in. stock removal. Inching of tool into bore, tool rotation, reciprocation speed, and tool withdrawal all controlled by one lever. Tool expanded by adjustable sleeve on head that does not rotate with tool. Micromatic Hone Corp., Detroit, Mich.

DIE FILING MACHINE. Has infinite speeds between 170 to 470 strokes per minute. Scotch yoke mechanism gives true up-and-down motion, runs in oil bath. Neoprene bellows seals dirt from mechanism. Dimensions: center of table to overarm, 8 in.; max height of work above table, 8 in.; table size, 10⅜ in. square; max tilt angle, 25 degrees both ways; stroke, 1½ in. Powered by 110-volt, 60-cycle, 1750-rpm motor. All American Tool & Mfg. Co., Chicago, Ill.

PLAIN GRINDING MACHINES. In 14 and 16-in. sizes. Grinding wheel spindles run on "Filmatic" bearings—multiple bearing segments carry spindle on wedge-shaped oil film. Grinding wheel stops if lubrication fails. Table traverse rates variable between 3 and 120 in. per minute through electronically-controlled d-c motor. Automatic acceleration and deceleration of table at reverse

Shaft Location With HYATTS





HERE is one of the ten basic types of Hyatt Hy-Load Cylindrical Roller Bearings, the "B-UL". It provides not only the high load capacity and long life of a cylindrical roller bearing but shaft location as well. It may be used on any application where two bearings are mounted opposed to locate shafts or gears in both directions.

The "B-UL" application illustrated is just one of the many ways in which the Hyatt Hy-Load Series Bearings can be used. Each of the ten different types of Hy-Loads have their special purposes and applications. Full information about all ten types of Hyatt Hy-Loads is in our catalog 547 . . . a complete engineering guide to radial bearing selection and use.

. . .

Write now for your copy. Hyatt Bearings Division, General Motors Corporation, Harrison, New Jersey.

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MADE TO ORDER:
SPIRAL BEVEL
STRAIGHT BEVEL
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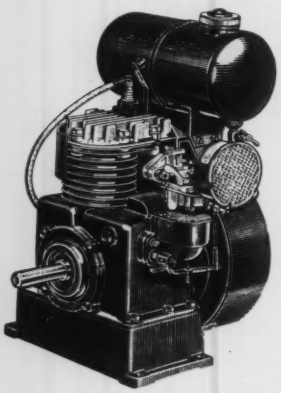
MASS production of fine gears is an accomplishment that requires the ultimate in manufacturing facilities and experience. And it's all here at Fairfield — metallurgical departments, batteries of the most modern machines, complete heat treating equipment, all operated by skilled craftsmen producing truly fine gears. Here is 30 years of experience translated into the gear-making "know-how" that is essential to manufacturing high quality, dependable gears.

Our facilities are complete for producing gears for a wide variety of applications. Why not consult Fairfield on your next gear problem? Your inquiry will receive prompt attention.

FAIRFIELD MANUFACTURING CO.
311 SO. EARL AVE. LAFAYETTE, INDIANA



Model K7-1, 1½ H. P. 4-cycle, single cylinder, air-cooled. Length 13", width 14", height 17", Weight 38 lbs.

POWER FOR:
Farm Equipment
Garden Tractors
Water Pumps
Concrete Mixers
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Rollers
Compressors
Sprayers
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KOHLER ENGINES
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PLUMBING FIXTURES • HEATING EQUIPMENT • ELECTRIC PLANTS • AIR-COOLED ENGINES

is provided. Headstock is a dead-spindle unit driven by d-c motor. Rheostat-controlled spindle speeds infinitely variable from 40 to 144 rpm on 14-in. machine, from 20 to 72 rpm on 16-in. model. Grinding wheel spindle driven by 20-hp motor, headstocks driven by 1½ and 2-hp motors, tables powered by 1-hp motors. Cincinnati Grinders Inc., Cincinnati, O.

PIPE THREADER. For threading ½ to 2-in. pipe and bolts. Four front jaws automatically grip pipe and 4 rear jaws center pipe to eliminate whipping. Chucks automatically operated by push-pull switch. Oil transmitted through telescoping lines directly onto cutting surfaces. Reamer attached to each head reams while machine is threading. Bell rings when proper length of thread has been cut. Quijada Tool Co. Inc., Los Angeles, Calif.

FLYWHEEL MACHINE. For drilling, chamfering and reaming up to 140 automotive flywheels per hour. Five-station index trunnion type unit uses fluid motor for indexing. Successive operations include drilling 16 holes, chamfering 16 and then 15 holes (in two operations), and reaming 3 holes. Except for loading and unloading, machine operation is automatic. The Cross Co., Detroit, Mich.

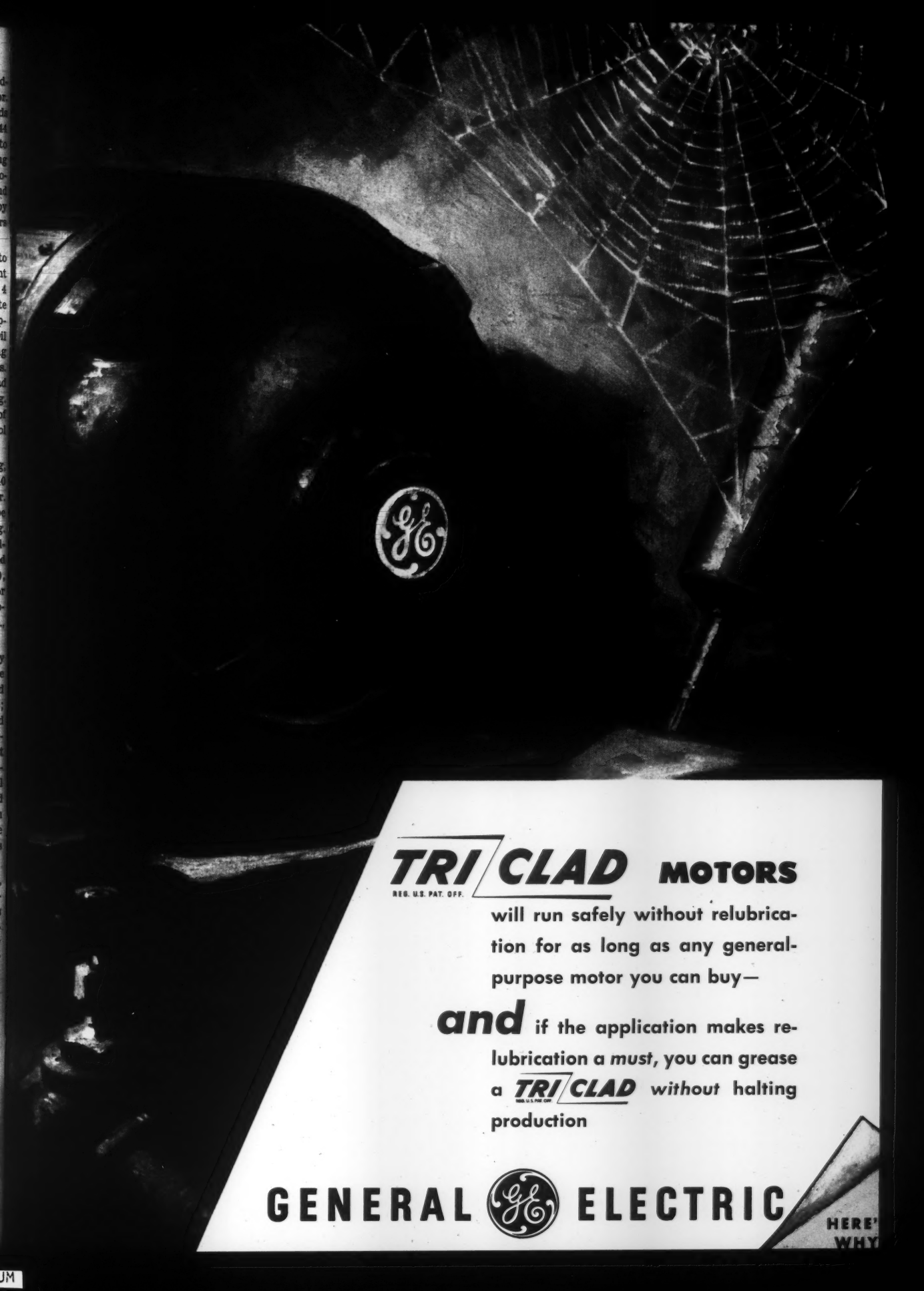
PORTABLE ELECTRIC DRILL. For body builders and body maintenance shops. Capacity, ¼-in. in metal and ½-in. in wood. Weight, 4¼ lb; overall length, 8¾ in.; no load speed, 1300 rpm. Includes pistol-grip handle and knob handle that rotates around tool to most comfortable position for operator. Drill has Jacobs 7B geared chuck and ball thrust bearing mounted on chuck spindle. Cummins Portable Tools, Div. of Cummins Business Machines Corp., Chicago, Ill.

PORTABLE ELECTRIC DRILL. Capacity, ¼-in. steel, ½-in. wood. Lightweight unit weighs 3 lb, measures 8¼ in. overall. Uses aluminum die-cast housing, 3-jaw Jacobs hex key chuck, trigger switch with locking device, and helical alloy steel gearing. No. 14 drill available for 115, 125 and 150 volts. No load speed, 1100 rpm. Stanley Electric Tools, New Britain, Conn.

Materials Handling

ELECTRIC INDUSTRIAL TRUCKS. Redesigned line of 12 Skylift trucks in capacities from 1000 to 4000 lb. Lift increased by 2 in. to 132 in. in all but one model; collapsed height remains at 83 in. All elec-

(*Continued on Page 281*)


TRI CLAD MOTORS
REG. U.S. PAT. OFF.
will run safely without relubrication for as long as any general-purpose motor you can buy—
and if the application makes relubrication a must, you can grease a TRI CLAD without halting production
GENERAL ELECTRIC
HERE'S WHY

TRI CLAD MOTORS will run safely without relubrication for as long as any general-purpose motor you can buy—

Tri-Clad extra lubrication "protection" can save you money because:

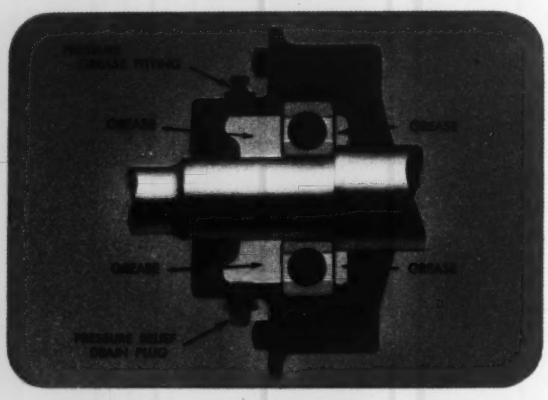
1. Tri-Clad's oversize grease reservoir and the heaviest standard-service bearings mean you do not have to bother with greasing between motor check-ups.

2. When relubrication is needed on those tough applications, you can grease a Tri-Clad without interrupting production-line operations.

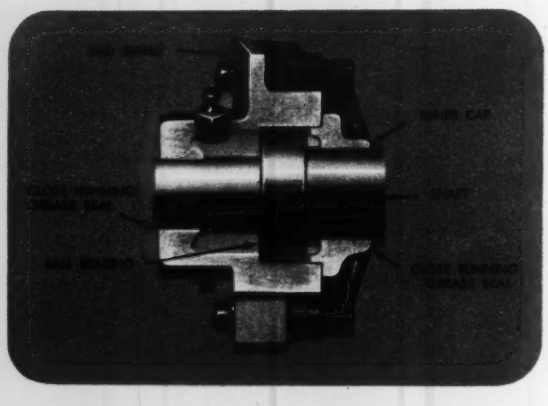
Tri-Clads are grease-gun easy to lubricate *on the job*. Moreover, a Tri-Clad motor will run safely where an ordinary motor would fail. Chances are you'll be spared the cost of a "special" motor.

YOU BE THE JUDGE! The best way to prove to yourself that Tri-Clad gives you the most for your motor dollar is to contact your local G-E office. Tri-Clad stocks are complete. *Apparatus Dept., General Electric Company, Schenectady 5, N. Y.*

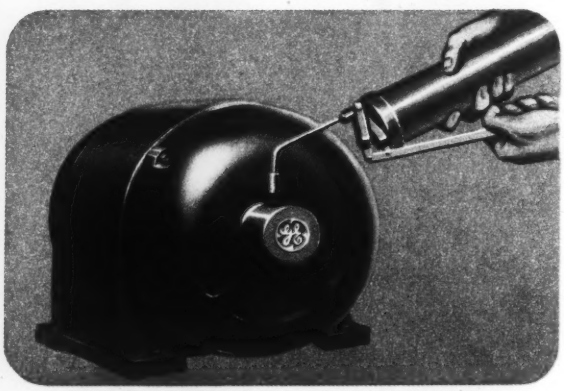
❶ **EXTRA BEARING PROTECTION** — Tri-Clad gives you extra bearing protection because heaviest standard-service bearings are carefully selected to withstand severe loads for long periods.



❷ **EXTRA GREASE** — Four times the ordinary amount of grease is packed into the large Tri-Clad grease reservoir. Since bearing life depends on grease, this means that Tri-Clad motors will run safely for years — for as long as any general-purpose motor you can buy.



❸ **SEALED-IN BEARINGS** — Bearings and grease are completely sealed in a cast housing with *long* running seals for extra protection from dirt, dust, and lubricant leakage.



❹ **PRESSURE-RELIEF GREASING** — An efficient system of pressure-relief lubrication (with standard fittings) enables a Tri-Clad motor to be quickly and easily greased on the job when and if it's needed.

GENERAL ELECTRIC

752-3

XUM

(Continued from Page 278)

trical controls placed in enclosed panel on dash, new controller and contactor make it impossible to reverse truck before coming to a stop. Tilt ram simplified with self-adjusting packing and heavier piston rod. Automatic Transportation Corp., Chicago, Ill.

ELECTRIC TIERING TRUCK. For use in restricted areas. Will handle 48 by 48-in. pallet loads at right angles from 6-ft aisle; right angle tiers 24 by 36-in. box skids from 5-ft aisle. In platform, fork and straddle models. Lyon-Raymond Corp., Greene, N. Y.

Plant Equipment

GASOLINE HAMMER ROCK DRILL. Self-rotating, hole-cleaning machine uses single-cylinder engine driving two pistons; one piston actuates flywheel magneto and cooling fan, other piston strikes directly on shank of drill. Spirally splined piston and ratchet used to rotate drill. Exhaust gas is forced through hollow drill to blow cuttings from hole to depth of 24 in. Hammer uses 1 by 4¼-in. shank hollow drill steels and detachable carbide-tipped drill bits from 1½ to 2-in. diameter. Syntron Co., Homer City, Pa.

DUST COLLECTOR. Exhausts air inside building, requires no piping. Air filtered through woven fabric and steel wool filter bags with heavy dust particles directed downward onto removable tray by baffle plate. Capacities: 235 cfm at 5400 fpm air velocity to 1025 cfm at 6000 fpm air velocity, in 4 sizes. The Hisey-Wolf Machine Co., Cincinnati, O.

Power Plant Equipment

DIESEL-ELECTRIC GENERATORS. Capacities of 5 models, 12½ to 55 kw. Engines started by cranking with 12-volt starting system; after less than one minute warmup, engine is shifted to full diesel operation by moving lever. Battery-recharging generator maintains starting batteries at full charge. All units equipped with voltage regulated type generators which may be paralleled. Standard equipment includes exhaust muffler, battery cables and terminal box. D. W. Onan & Sons Inc., Minneapolis, Minn.

GEARTURBINES. Industrial steam turbines with close-coupled, integral reduction gears for low-speed applications such as pumps, fans, compressors, etc. Units allow equipment to be run at proper speed while turbines are operated at most efficient speed. Single helical gears

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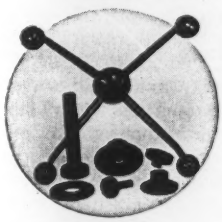
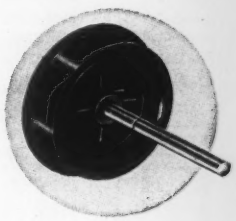
ECONOMICAL-ATTRACTIVE-PROTECTIVE CHEMICAL FINISHES

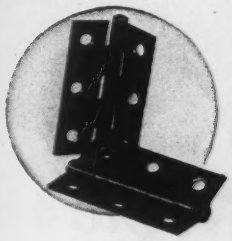
FOR FERROUS AND NON FERROUS METALS



STEEL



ZINC



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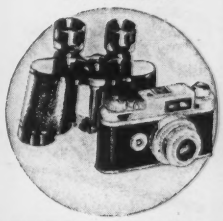
WRITE for "Black Book" of details on BLACK MAGIC. See the finish on your parts. Send samples for processing.



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Just because you used catsup on your steak, the waiter shouldn't assume you use it on the rest of your dinner too!

And just because one bearing is best lubricated by one particular grade of oil, you shouldn't assume that the same oil is best for *all* bearings on that machine. In many cases it isn't.

OIL CUPS permit you to lubricate each bearing with the oil best suited to that bearing—thus prolonging bearing life, reducing maintenance costs, cutting down-time, boosting production. And oil cups fortunately *cost very little.*

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